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AUTOMOBILE GAS TURBINE-OPTIMUM CYCLE SELECTION STUDY.

R. J. Rossbach

General Electric Company

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FINAL REPORT

AUTOMOBILE GAS TURBINE - OPTIMUM CYCLE SELECTION STUDY

Environmental Protection Agency
Contract No. 68-01-0406

Edited by
R. J. Rossbach

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June, 1972

—GENERAL ELECTRIC—
SPACE DIVISION
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A study was conducted with these objectives: to define the optimum gas turbine engine(s) capable of meeting the 1976 Federal Standards on automobile emissions and capable of being developed by the year 1975; to present recommendations for the development and demonstration of the selected optimum engine(s) by 1975 and for mass production of vehicles powered by such engines at earliest possible date after 1975. An analysis and preliminary design program was carried out in nine tasks. First ten gas-turbine engine types were screened by establishing the baseline technology, by carrying out a design Point Cycle Study and finally by making a preliminary selection of three engine types. Off design analysis, mission analysis, and economic analysis were performed on the two engines which were finally selected.

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1.0 SUMMARY

The requirements of the 1970 Clean Air Act Amendments are that the Environmental Protection Agency enforce stringent exhaust emission standards for new automobiles on unburned hydrocarbons and carbon monoxide in the 1975 model year and on nitrogen oxides in the 1976 model year. The 1976 standards are as follows:

Unburned hydrocarbons, grams/mile	0.41
Carbon monoxide, grams/mile	3.4
Nitrogen oxides, grams/mile	0.4

Although the automobile industry is working hard to meet the emission standards with the conventional automobile reciprocating engine, there are several other promising engine types which have a high potential of meeting the 1976 emission standards. The gas turbine is among the most important of these but the combustor for this type of engine does not yet meet the emission goals, nitrogen oxides being the most severe offender. In addition to the design compromises which may have to be made in gas turbine automobile engine in order to solve this problem, means must be found to mass produce the gas-turbine engine at low-cost and the fuel costs for the gas-turbine engine have to be made as low as practical.

The primary objectives of this contract are the following:

1. Define the optimum gas turbine engine(s) capable of meeting the 1976 Federal Standards on automobile emissions and capable of being developed by the year 1975.
2. Recommend to the Office of Air Projects, Advanced Automotive Power System Development Division the component research and development programs necessary to develop and demonstrate the selected optimum engine(s) by 1975 and to mass produce vehicles powered by such low emission engines at the earliest possible date after 1975.

An analysis and preliminary design program was carried out in nine tasks as shown schematically in Table 1, beginning with baseline technology or component state of the art, through a screening study of ten engines, a conceptual design study of two engines, a preliminary design study of two engines designed for low NO_x emissions, to a recommendation and program plan for engine development. A summary of the main engine performance parameters is listed in Table 2, for the engines that were studied in some detail. The first two columns refer to conceptual design engines that were selected in the screening study. The other columns refer to the preliminary design engines which were modified for lower NO_x emissions.

The preliminary design engines were based on two different approaches to reduce the NO_x emissions. PD-1A, a regenerative, single shaft engine with variable stators in the compressor and turbine was provided with a variable bypass around the regenerator for the combustor primary air. The GE low-NO_x combustor was one of the design features of this engine. This engine weighed 642 lbs. complete, had a design pressure ratio of 3.2:1 and it was found that this engine could meet the vehicle acceleration requirement with a rated power of 134 HP. The mission analysis indicated that the average fuel mileage was 13.59 mpg and that the required fuel cost for the life of the engine (10 years, 105,200 miles) was $2391. The economic analysis revealed that the engine first cost was $1539 and that the net cost of ownership for the engine and automobile was $11,776.

PD-2A, a recuperated, free turbine engine with a variable power turbine was designed so as to limit the combustor inlet temperature to 1000°F under all operating conditions. This engine weighed 613 lbs. complete, had a design pressure ratio of 6.6 and it was found that this engine could meet the vehicle acceleration requirements with a rated power of 134 HP. The mission analysis indicated that the average fuel mileage was 12.34 mpg and that the required fuel cost for the life of the engine (10 years, 105,200 miles) was $2635. The economic analysis revealed that the engine first cost was $1460 and that the net cost of ownership for the engine and automobile was $11,848.

The single-shaft engine, PD-1A, with the regenerator bypass for the primary combustor air was recommended for the advanced automobile gas turbine engine because the GE low-NO_x combustor, which has a higher

Table 1

Automobile Gas Turbine Optimum Cycle Selection Study

Task Breakdown

Task 1	Baseline Technology Component State of the Art
Task 2	Screening Study of Ten Engines Design Point & Preliminary Off-Design
Task 3	Select Three Engines for Conceptual Design
Task 6	Conceptual Design Layout
Task 4	Off-Design Performance
Task 5	Mission Analysis
Task 7	Economic Analysis
	Modify Two Designs for Low NO_x
Task 6	Preliminary Design Layout
Task 4	Off-Design Performance
Task 5	Mission Analysis
Task 7	Economic Analysis
Task 5	Mission Analysis of ICE and Two EPA Contractors Engines
Task 8	Recommended Configuration
Task 9	Program Plan

Table 2

Engine Design Point Parameters

Engine	CD-1	CD-2	PD-1	PD-2	PD-1A	PD-2A
Ambient temperature, °F	105	105	105	105	105	105
Inlet pressure, psia	14.553	14.553	14.553	14.553	14.626	14.626
Air flow rate, lb/sec	2.09	1.79	2.58	1.77	1.96	1.66
Output power, hp	150	150	150	150	150	150
Engine speeds, rpm	40K	55K/35K	32750	72K/50K	40000	80K/50K
Pressure ratio	3.6	5.0	3.2	8.0	3.2	6.6
Compressor efficiency	.80	.807	.80	.787	.823	.787
Cooling air fraction	.03	.03	.05	.05	.02	.03
Leakage fraction	.04	.05	.04	.03	.04	.01
Cold side H.E. pressure drop, %	1	1	.75	1	.75	1
Combustor pressure drop, %	4	4	4	4	4	4
Combustor efficiency	.99	.99	.99	.99	.99	.99
Turbine inlet temperature, °F	1900	1900	1850	1900	1900	1900
Turbine efficiency	.85	.84/.85	.85	.84/.85	.85	.84/.85
Accessory power, hp	4	4	4	4	4	4
Turbine exit pressure drop, %	2	2	3	2	1.5	1.5
H.E. effectiveness	.85	.85	.85	.78	.85	.73
Hot side H.E. pressure drop, %	7	7	7	7	4	7

potential of meeting the NO_x standard than the 1000°F combustor, is completely compatible with this engine and the net cost of ownership of this engine was no higher than for the free-turbine engine with the 1000°F combustor. The plan for the development and demonstration of this engine within 1975 was delineated, including component development, design, procurement and testing of the engine on dynamometers and in automobiles.

2.0 INTRODUCTION

The requirements of the 1970 Clean Air Act Amendments are that the Environmental Protection Agency enforce stringent exhaust emission standards for new automobiles on unburned hydrocarbons and carbon monoxide in the 1975 model year and on nitrogen oxides in the 1976 model year. The 1976 standards are as follows:

Unburned hydrocarbons, grams/mile	0.41
Carbon monoxide, grams/mile	3.4
Nitrogen oxides, grams/mile	0.4

Although the automobile industry is working hard to meet the emission standards with the conventional automobile reciprocating engine, there are several other promising engine types which have a high potential of meeting the 1976 emission standards. The most important of these are the Rankine engine, the stratified-charge engine and the gas-turbine engine. The present contract is concerned only with the gas-turbine engine.

With regard to the gas-turbine engine, there are several problem areas for which solutions must be found before this engine is suitable for application to the automobile. First, the gas turbine combustor does not yet meet the emission goals, nitrogen oxides being the most severe offender. Some design compromises may have to be made in gas turbine automobile engines in order to solve this problem. Combustor development is under way in several EPA contracts.

Second, it is desirable for the net cost of ownership of a gas-turbine-powered standard size six passenger sedan for the life of the vehicle to be within 10% of that for the same automobile with a reciprocating engine in 1970. This means the gas-turbine engine must be made by low-cost, production manufacturing techniques from economical materials.

Certain performance penalties will probably be associated with attempts to produce these engines economically.

Third, the fuel costs for the gas-turbine engine have to be as low as practical. This problem of fuel cost can be put into perspective by the realization that a gas-turbine engine must be rated at about 150 HP to give the automobile comparable performance to that of an automobile with a reciprocating engine; however, for most of its life, no more than 10 or 15 HP are required for most driving maneuvers, plus an average accessory load of 4 HP. In order to run economically at so low a percentage of rated power, the gas-turbine engine design must have some special design features.

The primary objectives of this contract are the following:

1. Define the optimum gas turbine engine(s) capable of meeting the 1976 Federal Standards on automobile emissions and capable of being developed by the year 1975.
2. Recommend to the OAP-AAPSDD the component research and development programs necessary to develop and demonstrate the selected optimum engine(s) by 1975 and to mass produce vehicles powered by such low emission engines at the earliest possible date after 1975.

Pursuant to the above objectives, an analysis and design program was carried out as shown schematically in Table 1. First, ten gas-turbine engine types were screened by establishing the Baseline Technology, by carrying out a Design Point Cycle Study and finally by making a Preliminary Selection of three engine types for design and analysis. Conceptual Designs were made for each of the three selected engine types. At this point it became apparent that the third of the three selected engines was too complicated compared to the small increase in performance and was dropped from further consideration on the basis of cost. Off Design Analysis, Mission Analysis and Economic Analysis were performed on the remaining two selected engines.

Although combustor developmental work is still going on, it became apparent that in order to meet the Federal Standards on NO_x emission the gas turbine cycle might have to be compromised (reference 1). As a result the two engines chosen for Preliminary Design were based upon the Conceptual Design engines but each was designed for a different approach to reducing the NO_x emissions. These Preliminary Designs were carried out in parallel with Off-Design Analysis, Mission Analysis and Economic Analysis, with the object being to identify an engine for powering an automobile with the least cost of ownership to the owner over a ten-year life during which the automobile will be driven 105,200 miles. For absolute comparison, two engines supplied by EPA contractors were also subjected to a Mission Analysis. Finally, Recommended Configurations and Program Plans for their development and demonstration were formulated.

3.0 RESULTS

3.1 Baseline Technology

3.1.1 Filter

A literature review (e.g., reference 2) indicated that air inlet filtration systems are usually designed for a pressure drop of four inches of water, or about one percent of the inlet pressure, and this value was used in the engine performance calculations.

3.1.2 Compressor

For the off design performance calculations, two General Electric generated compressor maps were selected and were fitted for the performance calculation program. One map for a design pressure ratio of 6, was scaled from 4 to 8, and the other map for a design pressure ratio of 3, was scaled from 3 to 4. The high pressure ratio map was the predicted performance for a developmental compressor and was based upon experimental data for a lower pressure ratio. The low pressure ratio map was obtained experimentally during a compressor development program. These maps are consistent with the literature on centrifugal compressors. Figure 1 of reference 3, for example, indicates that a polytropic efficiency slightly over 0.85 has been achieved for pressure ratios up to 6:1.

The compressor map for a low pressure ratio compressor with variable geometry is shown in Figure 1. The effect of variable geometry on compressor performance was taken from references 4 and 5. The compressor of reference 4 was designed for a pressure ratio of 2.3, and was tested with five diffuser vane angles of the same diffuser blade design. The results showed practically no decrease in either pressure ratio or efficiency as the diffuser area was reduced. Since the off design performance

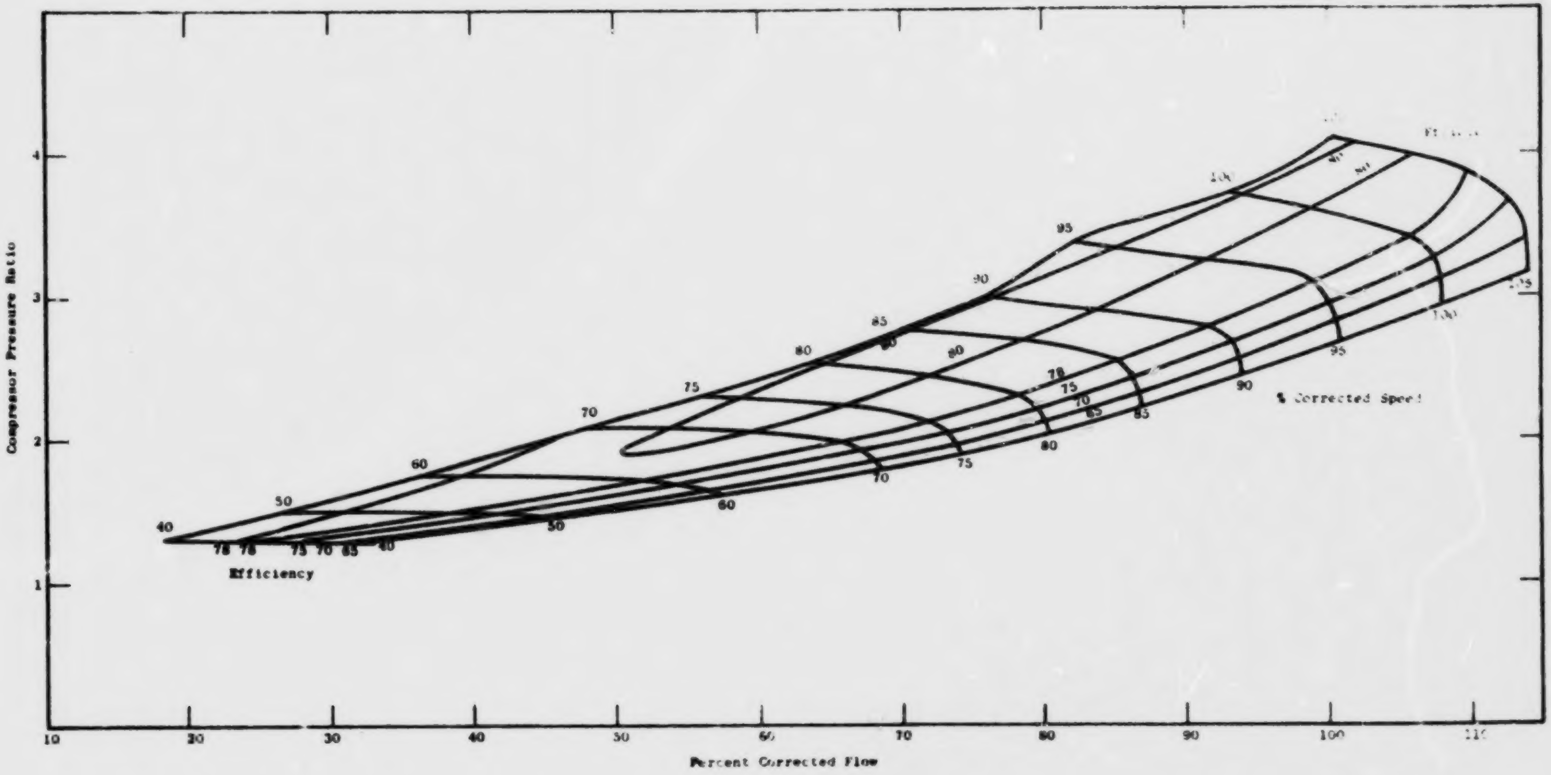


Figure 1. Compressor Map Design Pressure Ratio = 3.6.

of the automobile gas turbine engine is more important that the rated performance, which is only required for acceleration and maximum vehicle speed, the engine compressor would be designed for good performance at lower speeds and flows, with enough flow capacity at high speeds. The efficiency of the compressor has been reduced about three percent from the experimental data for this reason, and no additional penalty was taken for closing the diffuser to 45% of the maximum flow area. The flow variation as a function of corrected speed and diffuser area was taken from reference 5. The diffuser vanes would have about ten degrees of travel and it has been estimated that the movement could be achieved in less than 0.1 second.

The compressor map for high pressure ratio, fixed geometry compressors is shown in Figure 2. Since this compressor has fixed geometry, no efficiency penalty was taken.

3.1.3 Heat Exchanger

Ceramic Rotary Regenerator - A ceramic core for the regenerator was chosen over one of stainless steel because of its lower weight at the same thermal rating and pressure drop. The ceramic core is competitive because of its availability with small passage diameters and because of its favorable material properties of low density and high specific heat.

The Corning L689 regenerator core material was used in the design studies because of its compactness and availability. Reliable thermal-hydraulic data on this surface was obtained by A.L. London, et al., reference 6. The Fanning friction may be represented by

$$\frac{\Delta P}{4\,\frac{L}{d_h}\,\frac{G^2}{2\rho g_c}} \equiv f = 13/Re \qquad (1)$$

and the j-factor is given by:

$$\frac{h\,Pr^{2/3}}{C_p G} \equiv j = 3/Re \qquad (2)$$

where h is the heat transfer coefficient, ΔP is the pressure drop, G is the mass flux, Pr is the Prandtl number, L is the gas flow length, d_h is the hydraulic diameter, ρ is the density, C_p is the specific heat, μ is the viscosity, d_h is the hydraulic diameter, and g_c is the gravitational

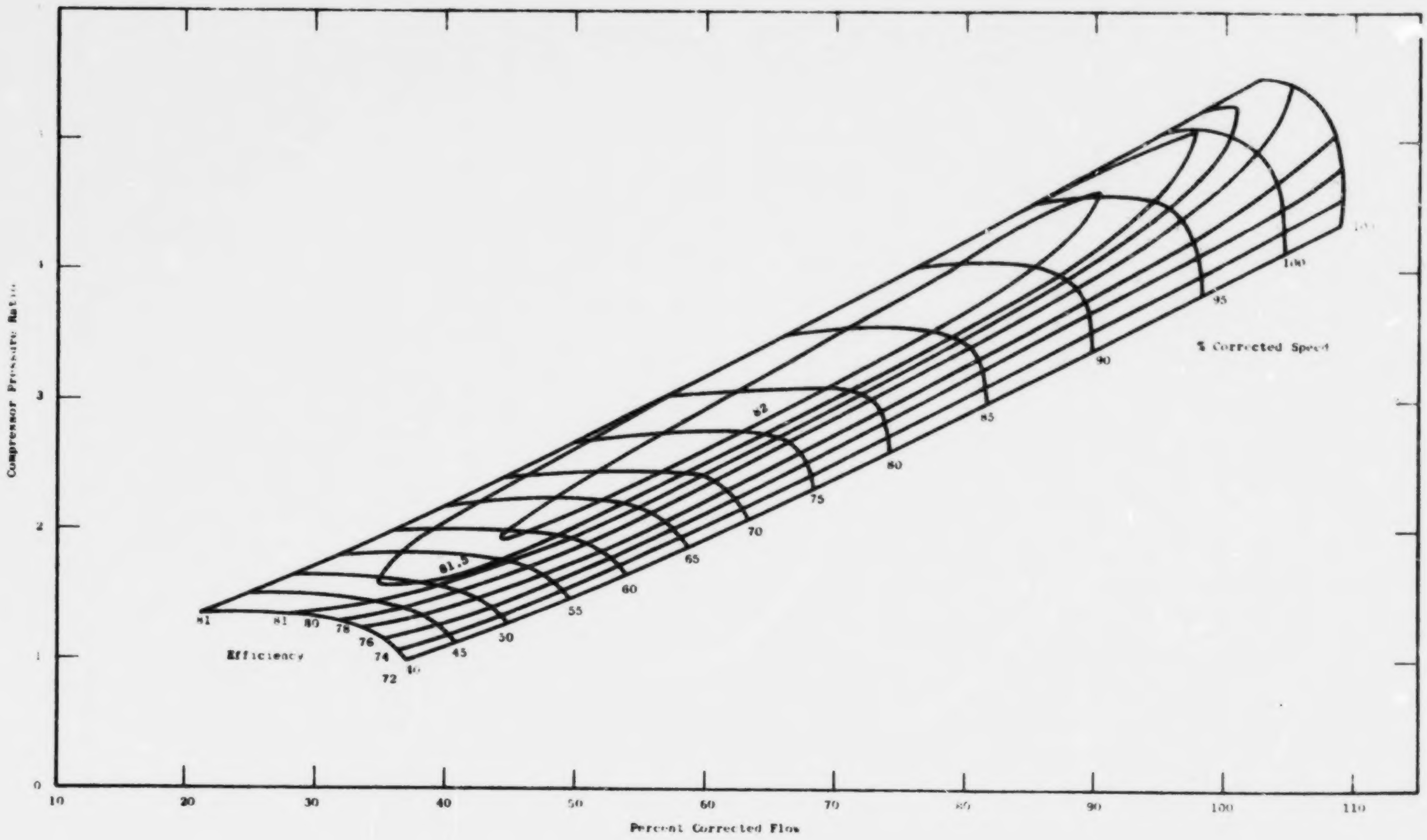


Figure 2. Compressor Map Design Pressure Ratio, 5.0.

conversion constant. The Reynolds number (Gd_h/μ), is based on passage hydraulic diameter and the data covers the range of $60<Re<900$.

The Corning L689 core has 1000 triangular holes per square inch, with an open base width of 0.038 inch and an open height of 0.020 inch; these dimensions correspond to a hydraulic diameter of 0.020 inch and a porosity of 70%.

Parametric design curves for rotary counterflow regenerators with a Corning L689 core are given in the Figures 3 and 4. Figure 3 shows the parametric effect of pressure ratio (and implicitly, the flow rate) upon the geometry and weight of a ceramic regenerator at fixed thermal effectiveness of 0.85 and total fraction pressure drop ($\Delta P/P$) of 0.08; the required flow rate varies with cycle pressure ratio and is based on cycle data. The required flow rate is lower at the higher cycle pressure ratios, and this dependence accounts for the effect of pressure ratio on diameter. The ratio of hot-gas flow area to cold-air flow area ($Ag/Aa = 1$) is taken as unity. The top rotational speed is 25 RPM, and there is a 4 inch diameter center hub for mounting and driving.

The curves in Figure 4 show the dependence of core geometry and weight on effectiveness for two values of total percent pressure drop. The pressure ratio is 3.2, and the turbine inlet temperature is 1900°F. As the required thermal effectiveness is reduced, the size and weight of the regenerator are reduced.

It is assumed that the regenerator is composed of two identical discs; if only one disc were desirable, then the disc diameter would be 41.4% larger than the two disc case, with the thickness unchanged for the same total percent pressure drop and thermal performance.

The off-design performance was determined as a function of the design point values in the following manner. Based on London's data for the j-factor in Eq. 2, the heat transfer coefficient is independent of flow rate. Then, it can be shown that the off-design number of transfer units, NTU, is related to the design-point value, NTU^* by:

$$\frac{NTU}{NTU^*} = \frac{\mu_a}{\mu_a^*} \frac{W^*}{W} \qquad (3)$$

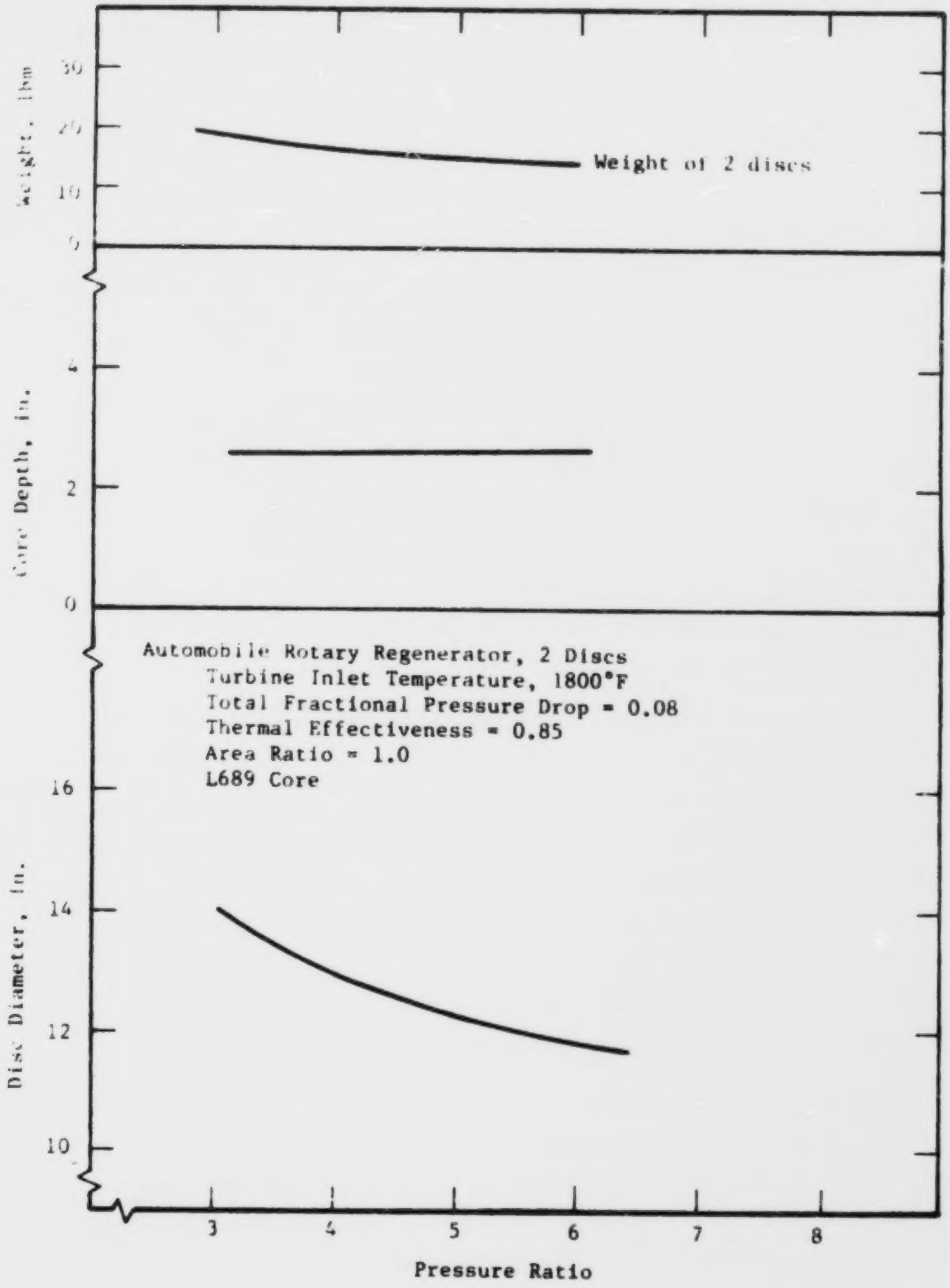


Figure 3. Effect of Pressure Ratio Upon Ceramic Regenerator Size and Weight.

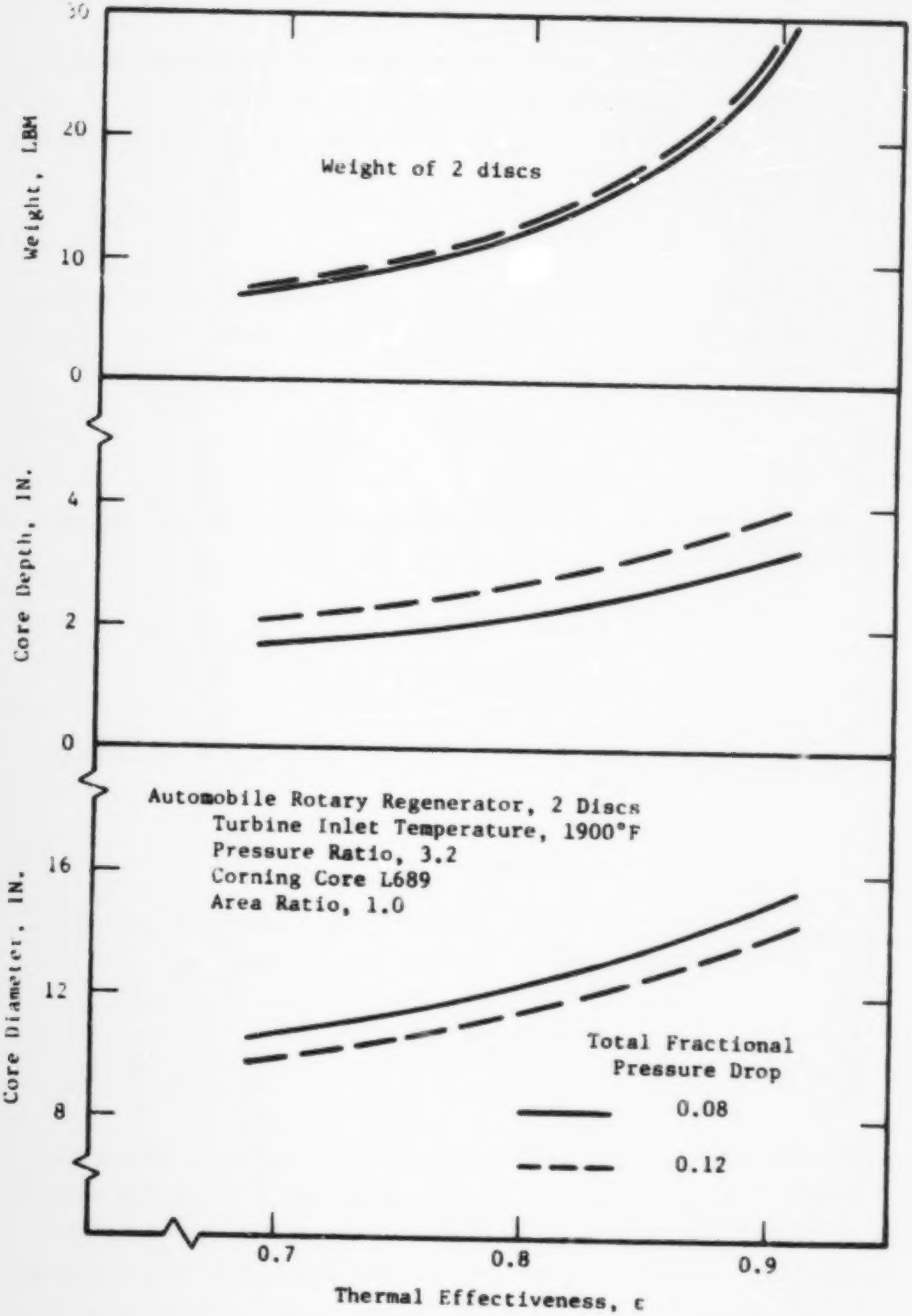


Figure 4. Effect of Thermal Effectiveness Upon Ceramic Regenerator Size and Weight.

where the star refers to the design-point values of air viscosity (μ_a) and flow rate (W). The assumptions made to deduce Equation 3 include:

1. Equation 2 is a valid representation of the thermal data.
2. Average thermophysical properties may be used.
3. The average specific heats of the hot gas and of the cold air are equal,

$$C_{pa} = C_{pg} \tag{4}$$

4. The ratio of the viscosity of the air and of the gas at the design-point is the same as that at off-design,

$$\frac{\mu_a}{\mu_g} = \frac{\mu_a^*}{\mu_g^*} \tag{5}$$

Then, to deduce the off-design value of the thermal effectiveness, ε, the value of NTU from Eq. 3 was used:

$$\varepsilon = \frac{NTU}{1 + NTU} \tag{6}$$

The assumptions used to obtain Equation 6 are:

1. The ratio of the capacity rates of the two streams is unity:

$$\frac{W_a\ C_{pa}}{W_g\ C_{pg}} = 1 \tag{7}$$

2. The ratio of the capacity rate of the regenerator matrix mass to the capacity rate of the air side is infinite:

$$\frac{\left.M\ C_p\ W_{rot}\right)_{matrix}}{W_a\ C_{pa}} = \infty \tag{8}$$

Neither Equation (7) nor (8) is strictly true; for example, the ratio of Eq. 7 is typically 0.9 for this application, while the value of ratio of Eq. 8 is typically 3. However, the effect of the two assumptions offset each other; the overall result obtained from Equation 6 at the off-design value is conservative and slightly low by about 1% at the low flow rates, as has been calculated at a typical off-design point.

For the off-design percent pressure drop on each side (ΔP/P), the following relation can be deduced:

$$\frac{(\Delta P/P)}{(\Delta P/P)^*} = \frac{W}{W^*}\frac{\mu}{\mu^*}\frac{T}{T^*}\left(\frac{P^*}{P}\right)^2 \qquad (9)$$

where T is the average fluid temperature, P is the pressure level, and W is the flow rate.

The assumptions made to deduce Equation (9) include:

1. Equation 1 is a valid representation of the hydraulic data. The main implication of Equation 1 is that pressure drop is linear with flow.
2. Average thermophysical properties may be used.
3. The densities of the air and gas vary as:

$$\rho \sim \frac{P}{T} \qquad (10)$$

The percent pressure drop of each side is calculated with Equation (9) and then the two values are summed for the total. The above equations are part of the engine off-design performance program and are used to calculate regenerator effectiveness and pressure drop at off-design conditions.

Ceramic Cross Flow Recuperator - In addition to the ceramic rotary regenerator discussed above, a static ceramic crossflow recuperator was chosen for one engine. In this application, the thermal effectiveness required of the exchanger was low (typically, 80%). A static recuperator has no leakage problem due to moving seals and no carryover losses. The crossflow arrangement also minimizes the manifolding with only a small penalty in thermal effectiveness at values of effectiveness in the range of 80%.

The Owens-Illinois "Cer-vit" ceramic core was examined as a possible crossflow recuperator. The configuration of the core consists of layers of continuous circular tubes of small diameter stacked with tube axes of alternate layers at 90° angles. Data to calculate thermal-hydraulic performance has been suggested by the manufacture. The Fanning friction factor for this surface was obtained from reference 7 and can be represented as:

$$f = 20.8/\mathrm{Re}^{0.947} \tag{11}$$

The j-factor for heat transfer was obtained from reference 8 and was taken as:

$$j = 4/\mathrm{Re} \tag{12}$$

The relationship for thermal effectiveness, ε, of a single-pass crossflow heat exchanger with both fluids unmixed can be obtained with a numerical integration of a series solution developed by Mason, reference 9:

$$\varepsilon = \mathrm{NTU}\ \phi \tag{13}$$

where

$$\mathrm{NTU} \equiv \frac{UA}{\left(W\ C_p\right)_{min}} \tag{14}$$

$$\phi \equiv \frac{1}{(\mathrm{NTU})^2\ \mathrm{CR}} \sum_{n=0}^{j \to \infty} R_n\ (\mathrm{NTU})\ R_n\ (\mathrm{CR\ NTU}) \tag{15}$$

$$R_n(w) \equiv 1 - e^{-w} \sum_{k=0}^{n} \frac{w^k}{k!} \tag{16}$$

w = dummy summation variable

k,j,n = integers

$$\mathrm{CR} \equiv \frac{(WC_p)_{min}}{(WC_p)_{max}} = \text{circulation ratio} \tag{17}$$

WC_p is the capacity rate, with "min" and "max" denoting the streams with the larger and the smaller values of capacity rate.

The results for thermal effectiveness ε are obtained as a function of number of heat transfer units NTU and circulation ratio CR by numerically summing Equations (15) and (16); the results are shown in Figure 5.

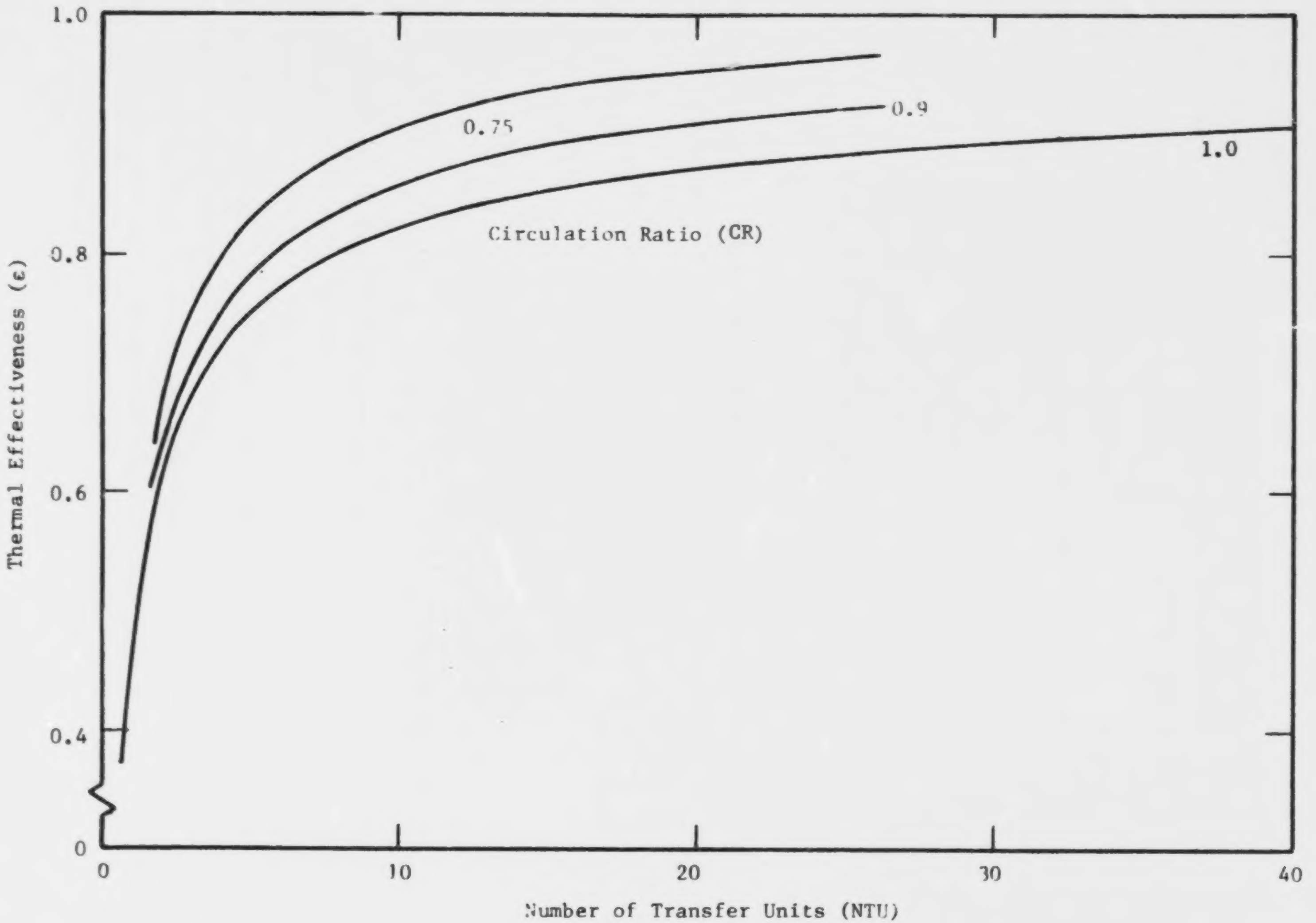


Figure 5. Effectiveness NTU Relationship of Single-Pass Crossflow Heat Exchanger with Both Fluids Unmixed.

Then, with the thermal-hydraulic data of Equations (11) and (12) along with the effectiveness relationship of Figure 5, parametric design curves for static crossflow recuperators can be prepared. A typical design curve is given in Figure 6, where the hydraulic diameter of the matrix is the independent variable. The hot-gas flow length was set at 6 inches for packaging, and the aspect ratio of the hot-side face area was set at 2:1; two regenerator exchangers of the indicated size are used.

The off-design performance of the recuperator is obtained as follows. Based on the j-factor of Equation (12), the heat transfer coefficient is independent of flow rate. Then, the off-design number of transfer units, NTU, is related to the design point value, NTU^*, by Equation (3) as given earlier:

$$\frac{NTU}{NTU^*} = \frac{\mu_a}{\mu_a^*} \frac{W^*}{W} \qquad (3)$$

Then, the off-design value of thermal effectiveness, ε, is obtained by going into Figure 5 with the off-design value of NTU and CR.

The off-design value of percent pressure drop on each side ($\Delta P/P$) can be deduced as:

$$\frac{(\Delta P/P)}{(\Delta P/P)^*} = \left(\frac{W}{W^*}\right)^{1.055} \left(\frac{\mu}{\mu^*}\right)^{0.945} \left(\frac{T}{T^*}\right) \left(\frac{P^*}{P}\right)^2 \qquad (18)$$

3.1.4 Combustor

The two parameters of interest to the engine designer are pressure drop and combustion efficiency. A review of the literature indicates that four percent pressure drop is feasible at the design point or maximum power condition. (For example, see reference 10). This value was used for the engine performance calculations and varied with Mach number for off design conditions. A combustion efficiency of 0.99 was used for all performance calculations, since this seems to be about the lowest value acceptable for a low emission combustor.

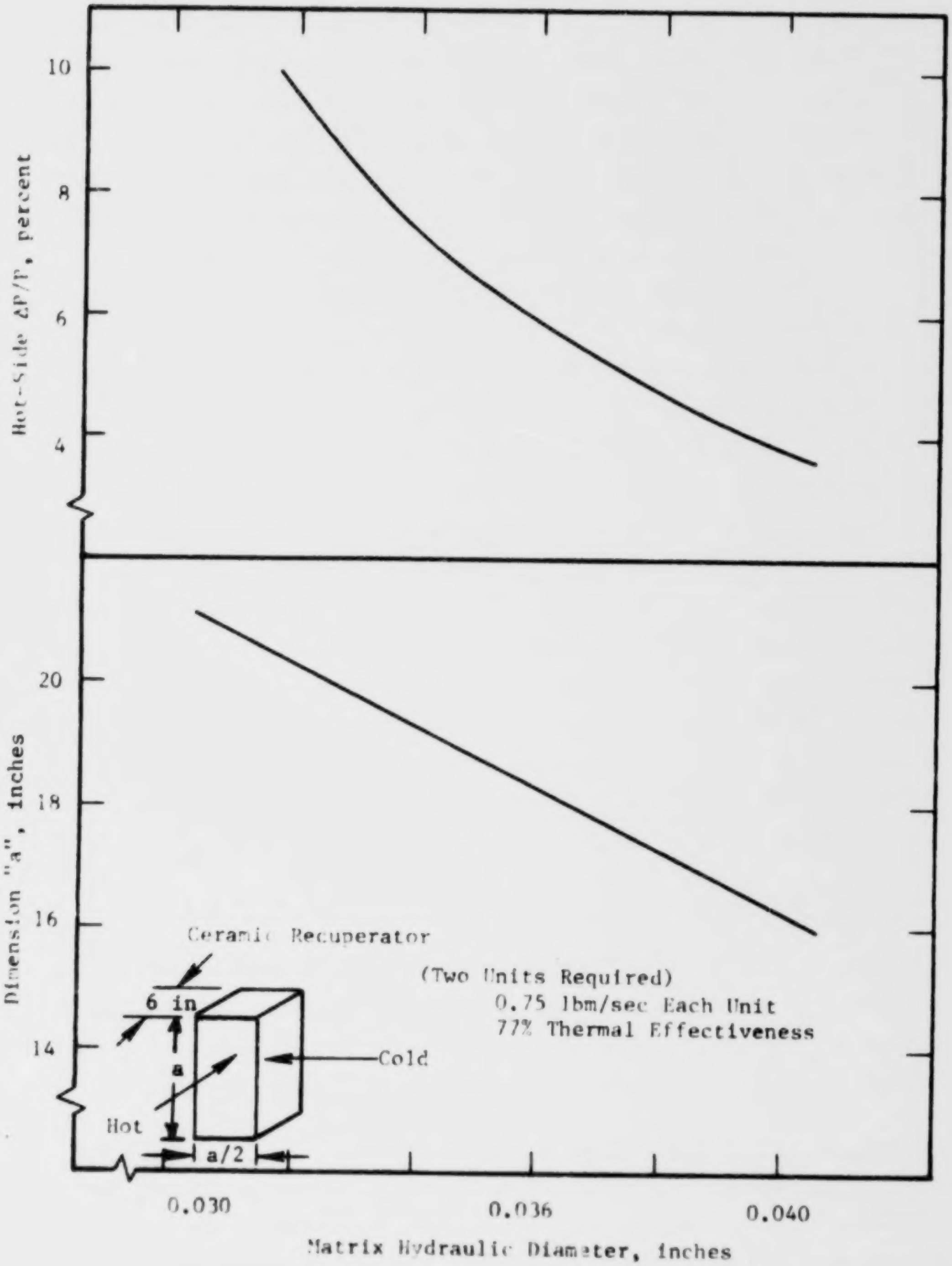


Figure 6. Effect of Matrix Hydraulic Diameter on Recuperator Geometry and Pressure Drop.

3.1.5 Turbine

The state of the art for axial flow turbines is given in reference 11. Turbine stage efficiency levels over 0.9 are shown for flow coefficients less than 1.0 and work coefficients less than 2. These data, for zero tip clearance, have been used by other authors, references 12 and 13, to develop and compare analytical loss calculation models. For the screening studies, turbine maps for one and two stage turbines with variable geometry were taken from reference 14 for use in the off design calculation programs. A method of describing turbine performance has been developed in Aircraft Engine Group (AEG) which permits reduction of turbine maps to essentially two single line plots, one for flow and one for efficiency. This method has been adopted for the off design calculation program and includes the effects of variable nozzle vanes on turbine performance. Shown in Figures 7 and 8 are turbine performance maps for the open stator position and for the closed position where the maximum flow function is 50% of the open position value. The speeds, efficiencies and flow rates are scaled to match the requirements of the various engine cycles. These maps are for a jet engine turbine and the turbine efficiency was scaled down to 0.85 at the match point reflecting the experience of AEG with small turbines. The major factor which causes the efficiency reduction is the small blade height for the automotive application. The effect of variable geometry is seen in Figure 8 to be a reduction in flow rate and a shifting of the peak efficiency to a higher corrected speed.

3.1.6 Transmission

Among the gas turbine engines under study, two differing requirements exist for transmissions. The conventional automobile automatic transmission with finite gear ratios and the infinitely variable transmission are the two types best fulfilling the differing requirements.

Conventional Automatic Transmission - Because the free-turbine engine has no mechanical coupling between the gas-generator shaft and the power-turbine shaft, the engine can remain coupled to the driving wheels at zero vehicle speed. Under these operating conditions, the power-turbine stator vanes are set to nullify positive torque on the

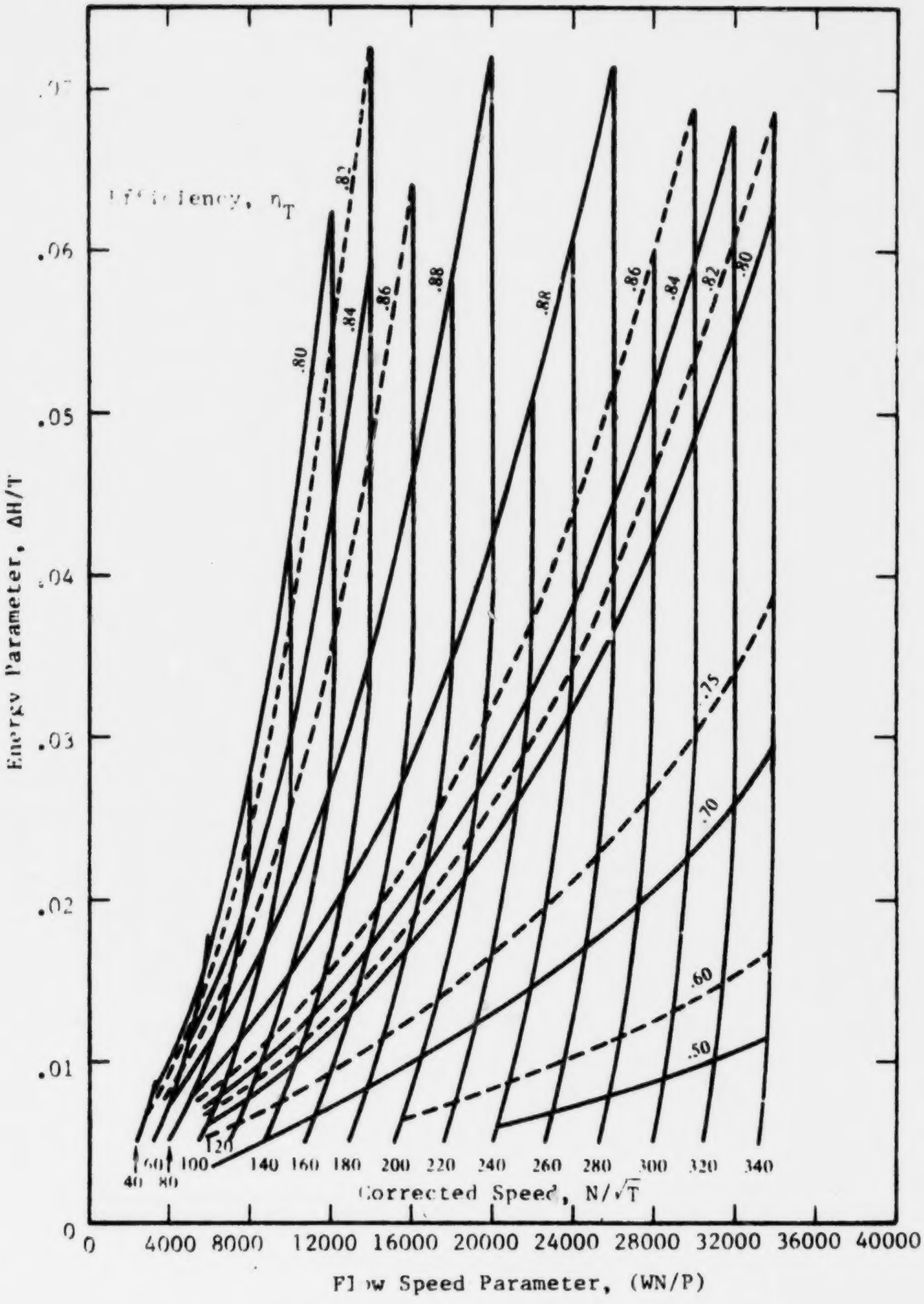


Figure 7. Turbine Map 100% Flow Function.

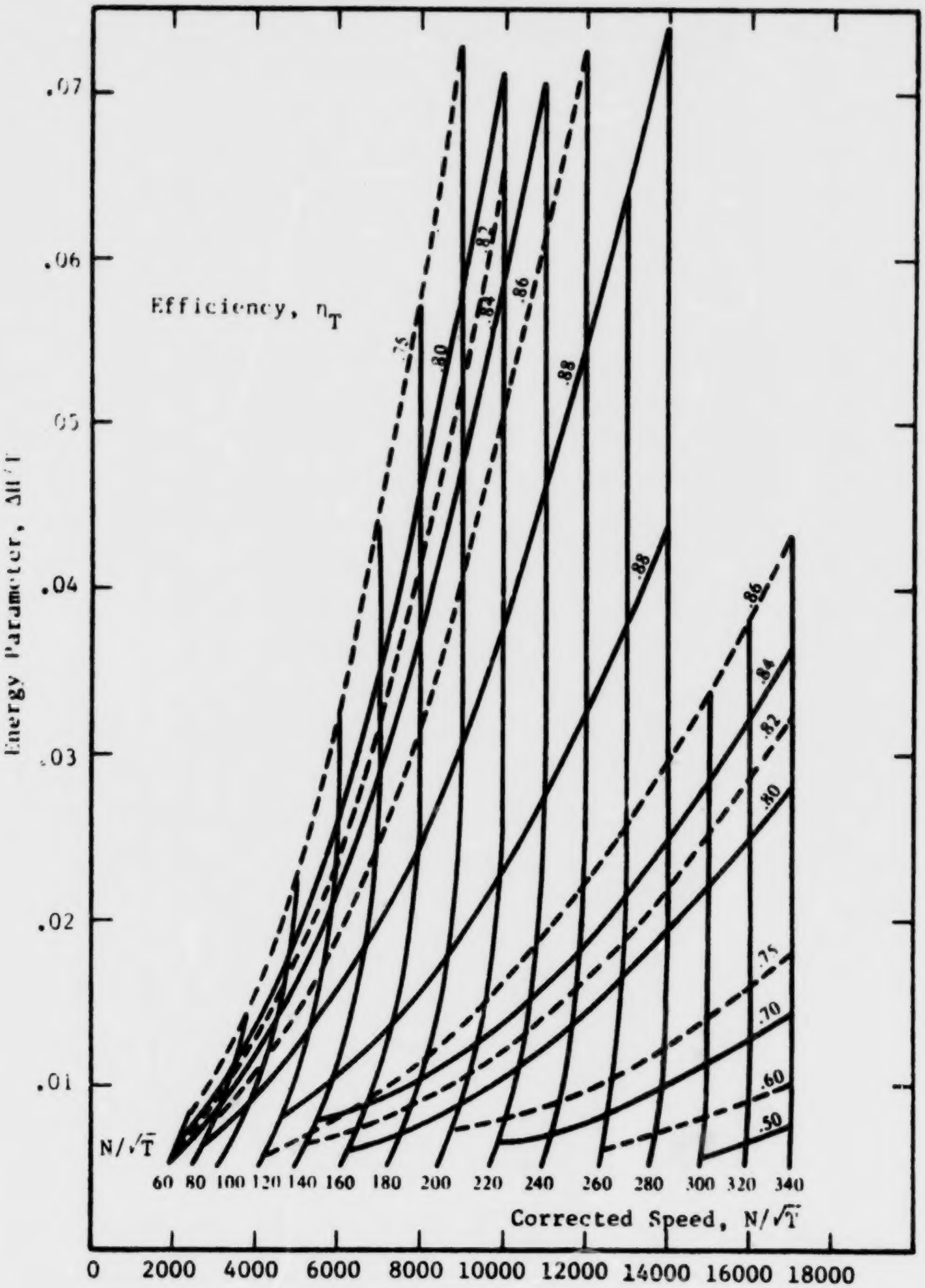


Figure 8. Turbine Map 50% Flow Function.

power turbine. The free-turbine engine develops full torque at zero vehicle speed. Thus, a conventional automatic transmission with three or four forward speeds is well suited to the engine. The free turbine can act as the coupling, eliminating the need for a torque converter. However, if the accessories are put on the free turbine shaft, this shaft may not be stopped and a torque converter is required.

For drive systems which require a torque converter, techniques and data given by Upton (reference 15) were used to size and select a torque converter. The torque ratio (T_{OUT}/T_{IN}) and the capacity factor ($K \equiv N_{IN}/\sqrt{T_{IN}}$, where N_{IN} is the input rotative speed of a torque converter) sized for one of the engines discussed below are shown in Figures 9 and 10. For converters of various sizes, the torque ratio as a function of speed ratio remains unchanged; however, the capacity factor (K) can be varied by the laws of dynamic similitude (reference 15) in order to match the torque converter to the particular engine. The results of such a match are given in Figure 11, which shows the torque-speed characteristic resulting from the data of Figures 9 and 10; the engine is geared so that the input speed to the torque converter is a maximum of 4000 RPM. Also, plotted in Figure 11 is the wide-open-throttle (WOT) torque of the engine. By scaling the capacity factor (K) of the converter according to the laws of similitude, a value of converter speed ratio of 0.9 is selected to fall on the 100% input speed (4000 RPM). This match of torque converter and engine results in good coverage of the engine's torque map by the torque converter characteristic.

During most of the time the automobile is operating, the transmission should shift to the gear ratio which allows the engine to operate closest to minimum fuel consumption. Under these conditions, the turbine inlet temperature will be less than or equal to the maximum continuous value of 1700°F. For starting up, merging with traffic, passing, etc., the turbine inlet temperature will be less than or equal to the maximum value of 1900°F. Under these conditions, high torque is more important than SFC. The shift points on the transmission must be set to accommodate both the economical and the high-torque operating conditions.

Data for a three speed automatic transmission from reference 16 were considered to be typical of conventional automatic transmissions.

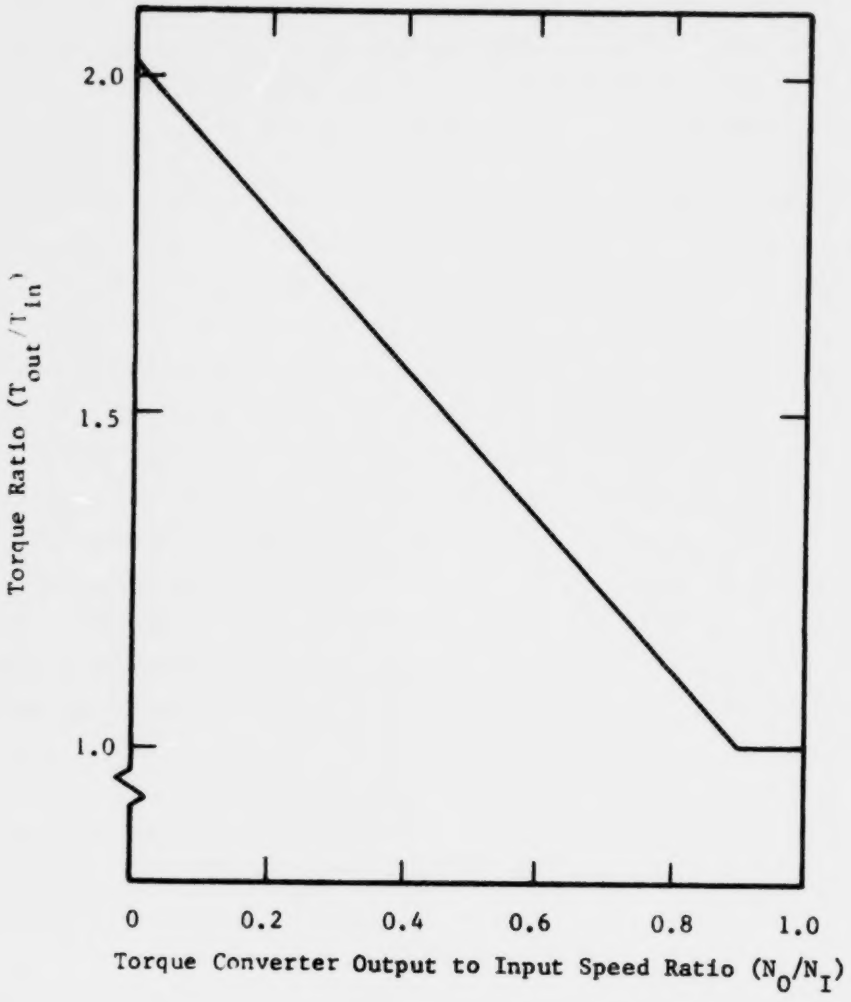


Figure 9 Torque Converter Torque Multiplication.

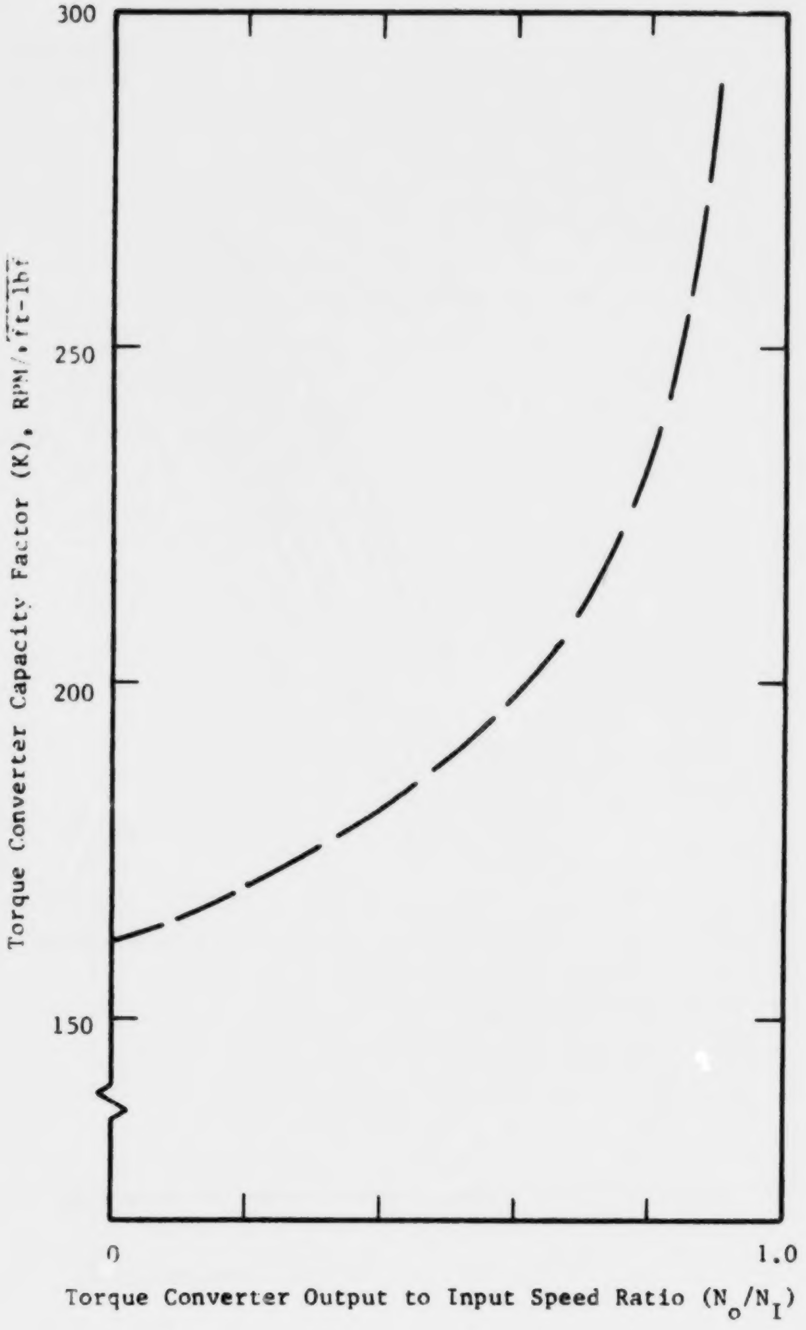


Figure 10. Torque Converter Capacity Factor.

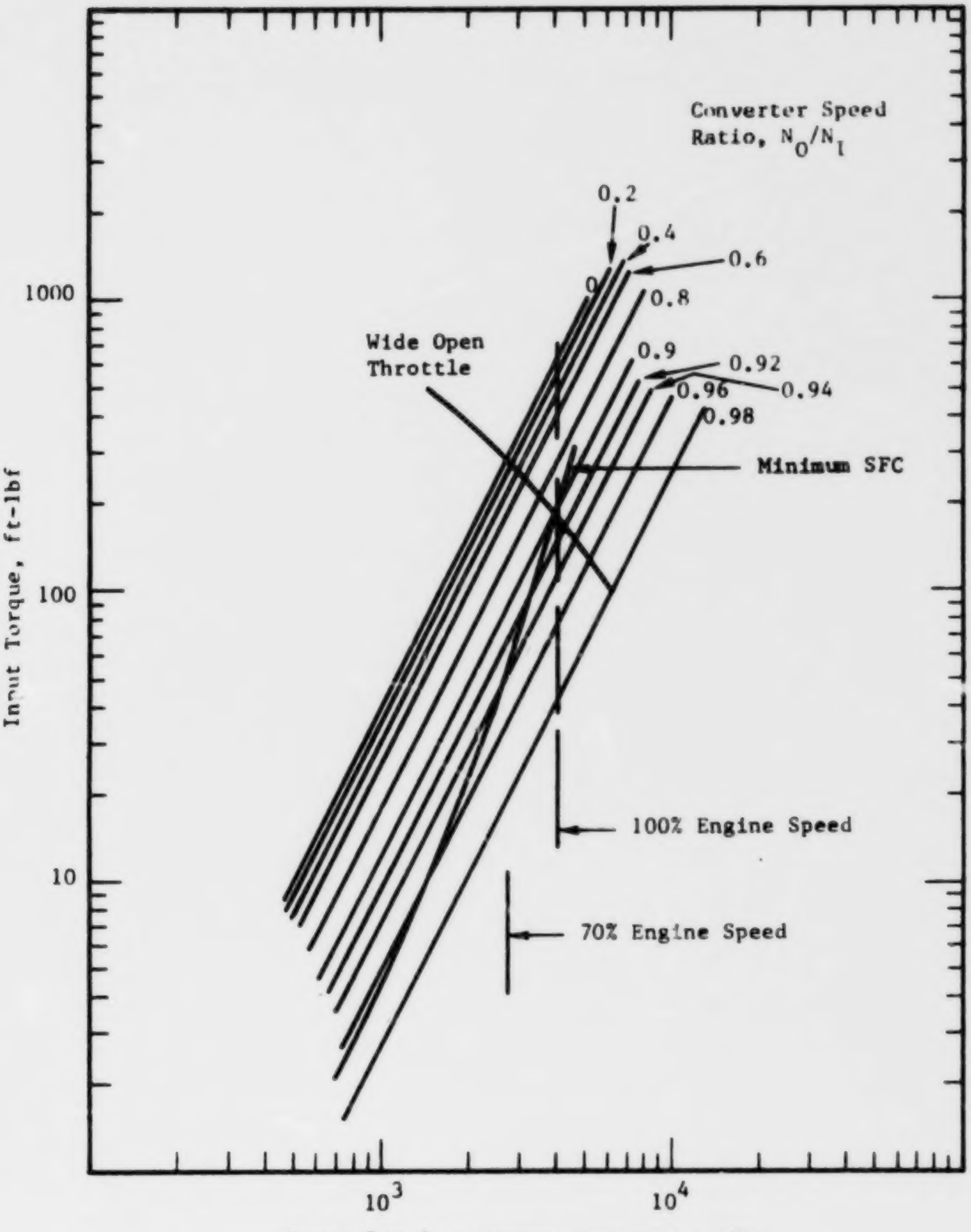


Figure 11. Match of Torque Converter with a Free Turbine Engine.

The transmission efficiency values were 0.90, 0.93 and 0.96 with gear ratios of 2.45, 1.45, and 1.00 for first, second, and third speeds, respectively.

Infinitely Variable Transmissions - Two infinitely variable transmissions have been under consideration. The first is the General Electric hydromechanical transmission model ALT10-C01 with a displacement of 11 in^3/rev. The transmission has a rated input speed and torque of 3000 RPM and 250 lb-ft, respectively. The output torque is limited to 880 lb-ft. The forward and reverse speed ratios vary from neutral to 0.8:1 and neutral to 2.5:1, respectively. The speed ratio is automatically controlled to the best value for the engine and load conditions. Because of the continuous speed variation through neutral, full engine dynamic braking is possible. Shown in Figure 12 is the variation of the transmission efficiency with output speed for various input horsepower values; the efficiency is independent of input speed (reference 17). The maximum transmission efficiency is 0.852.

The second infinitely variable transmission under consideration is the Tracor, Inc. toroidal transmission rated at 150 input horsepower. The transmission has an input speed of 10,000 RPM at high torque conditions and is capable of a variation in gear ratio over a range of 3:1 down to 1:15. The transmission has no neutral position and, therefore, requires the disengagement of the engine from the wheels at low vehicle speeds (less than 10 mph). A reversing gear is also required for this transmission. Shown in Figure 13 is the variation of transmission efficiency with input power for three output/input speed ratios (reference 18). The drive train would in addition include the efficiency of the slipping clutch at speeds less than 10 mph.

Shown in Figure 14 is a comparison of the transmission efficiency at vehicle crusing conditions. The low efficiency of the GE hydromechanical transmission significantly increases the fuel consumption. For this reason, the toroidal transmission was used with the single shaft engine in spite of the added complexity and reduced low-speed efficiency incurred by the slipping clutch which is required.

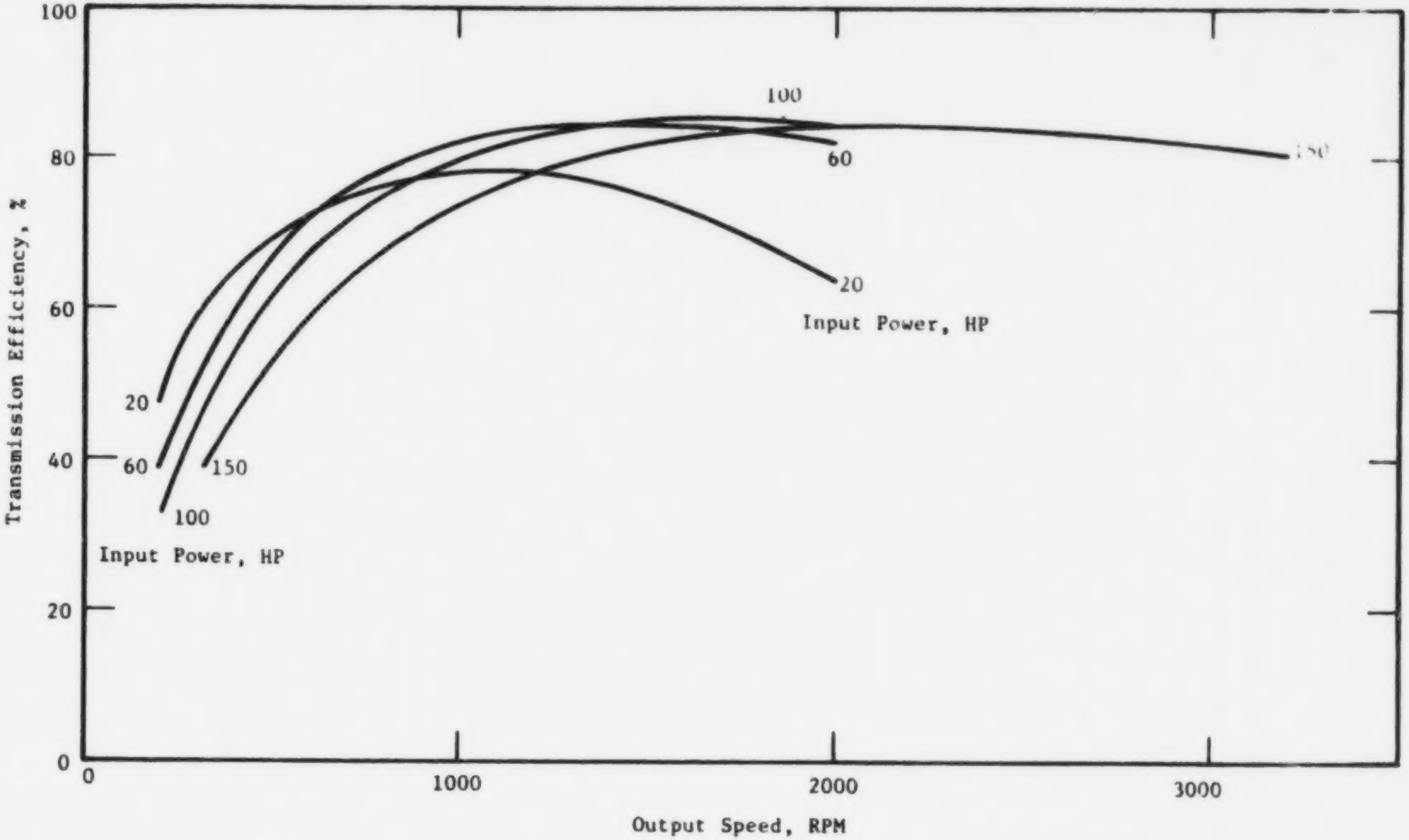


Figure 12. Variation of Efficiency of General Electric Co. Hydromechanical Transmission, Rated Power, 150 HP.

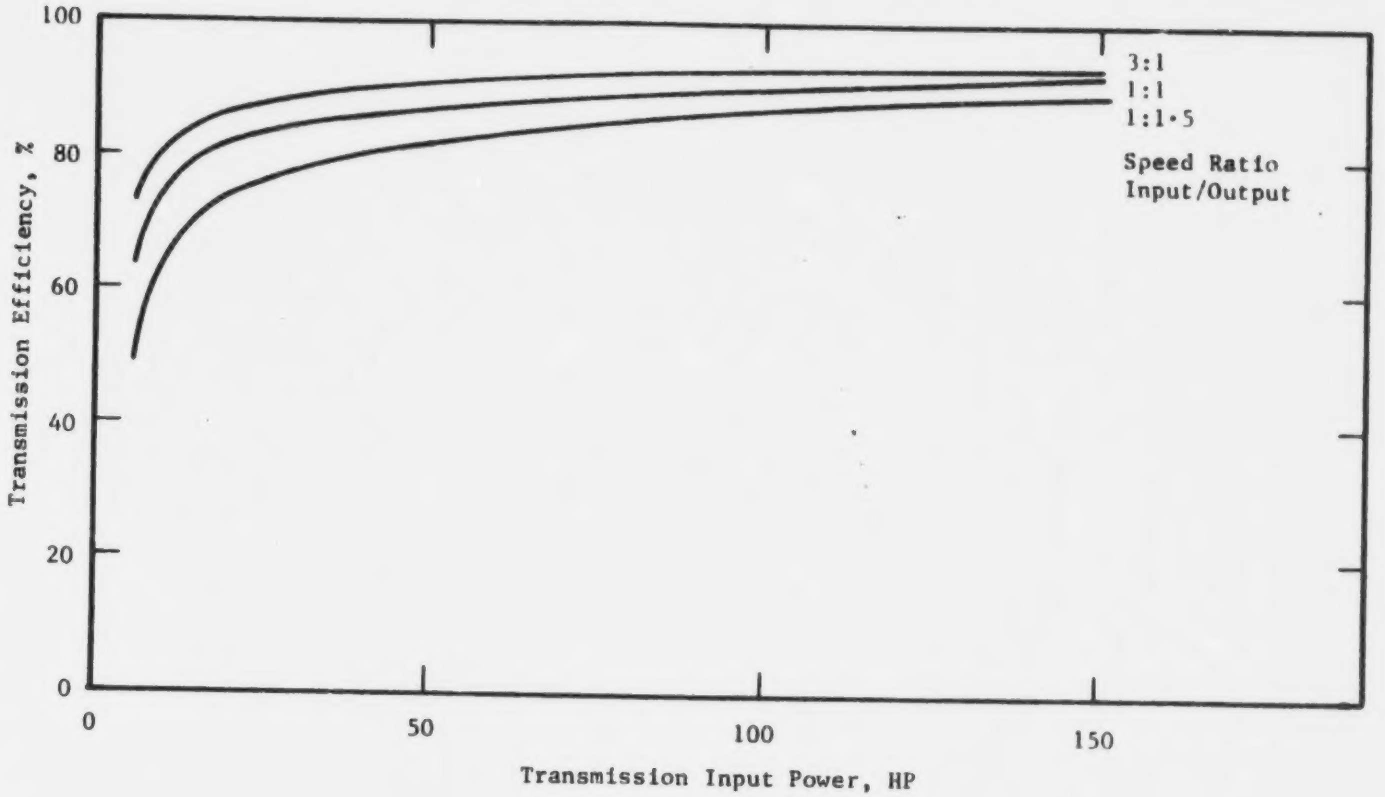


Figure 13. Variation of Efficiency of Tracor, Inc. Infinitely Variable Transmission, Rated Power, 150 HP.

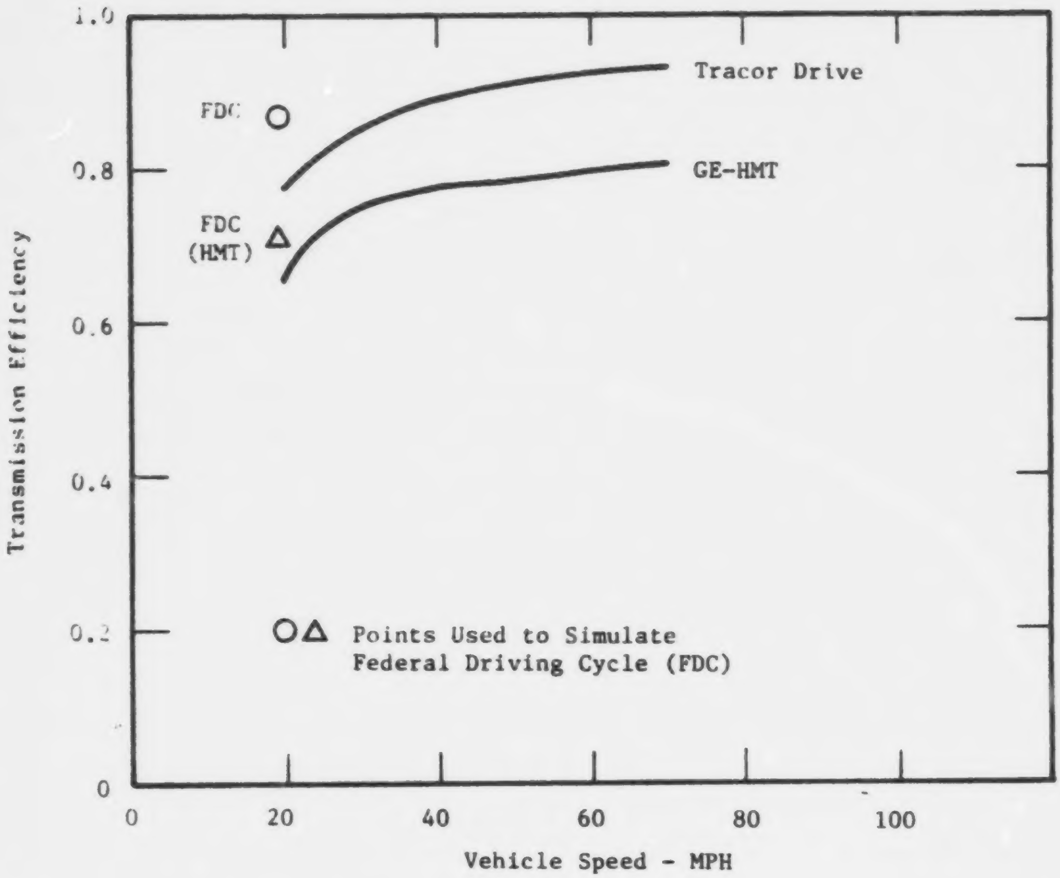


Figure 14. Transmission Efficiency at Cruising Conditions, Single Shaft Engine.

3.2 Design Point Cycle Study

The design point programs were written for use on the time sharing terminal, one for a single shaft engine, one for a free power turbine configuration and one for a two spool design. These programs accept inputs of compressor pressure ratio, temperatures, component efficiencies and pressure losses and calculate power output and specific fuel consumption. Since it was realized that the automobile gas turbine performance at low power levels is as important as performance at the design point, additional design parameters were calculated for use in the engine selection process. These included compressor and turbine wheel speeds, rotative speeds, turbine root centrifugal stresses, and gas temperatures relative to the turbine blades.

Design point calculations were made for all three kinds of engines, varying compressor pressure ratio, turbine inlet temperature, regenerator effectiveness and cooling flow. The inputs for the free turbine engine are shown in Table 3. The compressor efficiency corresponds to a polytropic efficiency of 0.85. Points were calculated for six pressure ratios, four turbine inlet temperatures, and two values of regenerator effectiveness. The compressor efficiency was varied with pressure ratio and the cooling flow was varied with turbine inlet temperature.

The variation of compressor tip speed with pressure ratio is shown in Figure 15. The symbols marked CD-2 and -3 indicate preliminary engine selections, the lower pressure ratio to have variable geometry. A major factor in the engine selections was the objective of minimizing stresses so that low cost, low alloy materials could be used. Low compressor tip speeds are consistent with low compressor stresses. The rotative speeds are shown in Figure 16. The gas generator speed is determined by the compressor pressure ratio and specific speed, and the power turbine speed is determined by the assumptions of turbine loading and diameter ratio. The turbine pitchline wheel speeds are shown in Figure 17 for the assumed loading. The pitchline wheel speeds of the gas generator turbine are slightly greater at higher turbine inlet temperatures because of the assumption of greater cooling flows. The wheel speeds of the power turbine are greater at higher turbine inlet temperatures because the specific work output (BTU/lb.) is greater. The root centrifugal stresses in the gas

Table 3

Free Turbine Engine

Input Variables:

Compressor pressure ratio	3	4	5	6	7	8
Compressor efficiency	.827	.821	.817	.813	.807	.80
Turbine efficiency		0.85				
Turbine inlet temperature		1700°F,	1800°F,	1900°F,	2000°F	
Cooling flow, %		1,	3,	5,	7	
Compressor inlet temperature		85°F				
Regenerator effectiveness		.9, .7				
Regenerator pressure drop		1% cold side, 3.5% hot side				
Combustor pressure drop		3%				
Tailpipe		1%				
Inlet pressure drop		1%				

Assumptions:

Compressor specific speed 70

Gas generator turbine exit Mach No. .5

Power turbine exit Mach No. .25

$$\frac{\text{Power turbine pitch dia}}{\text{Gas generator turbine pitch dia.}} = 1.5$$

Turbine loading, $g \frac{J\Delta h}{U_p^2} = 1.5$

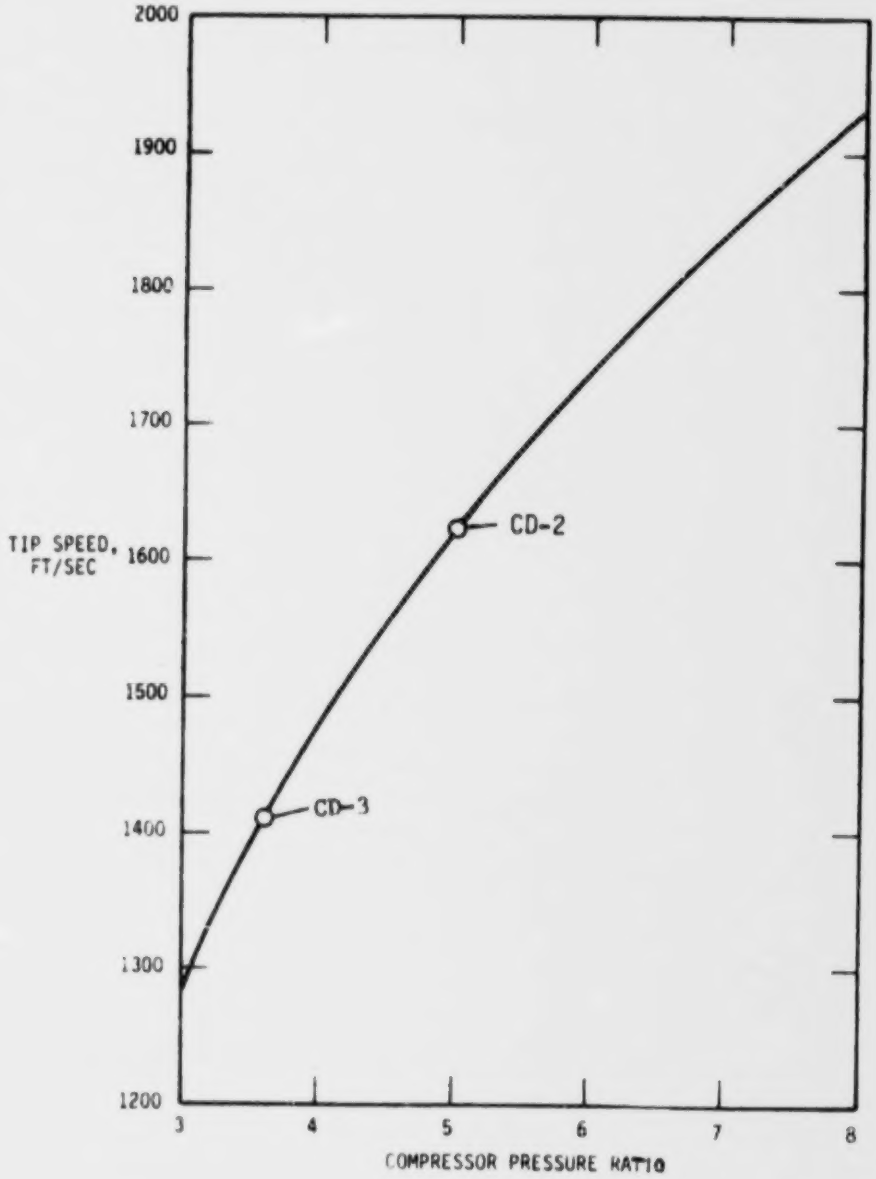


Figure 15. Free Turbine Engine Compressor Tip Speed.

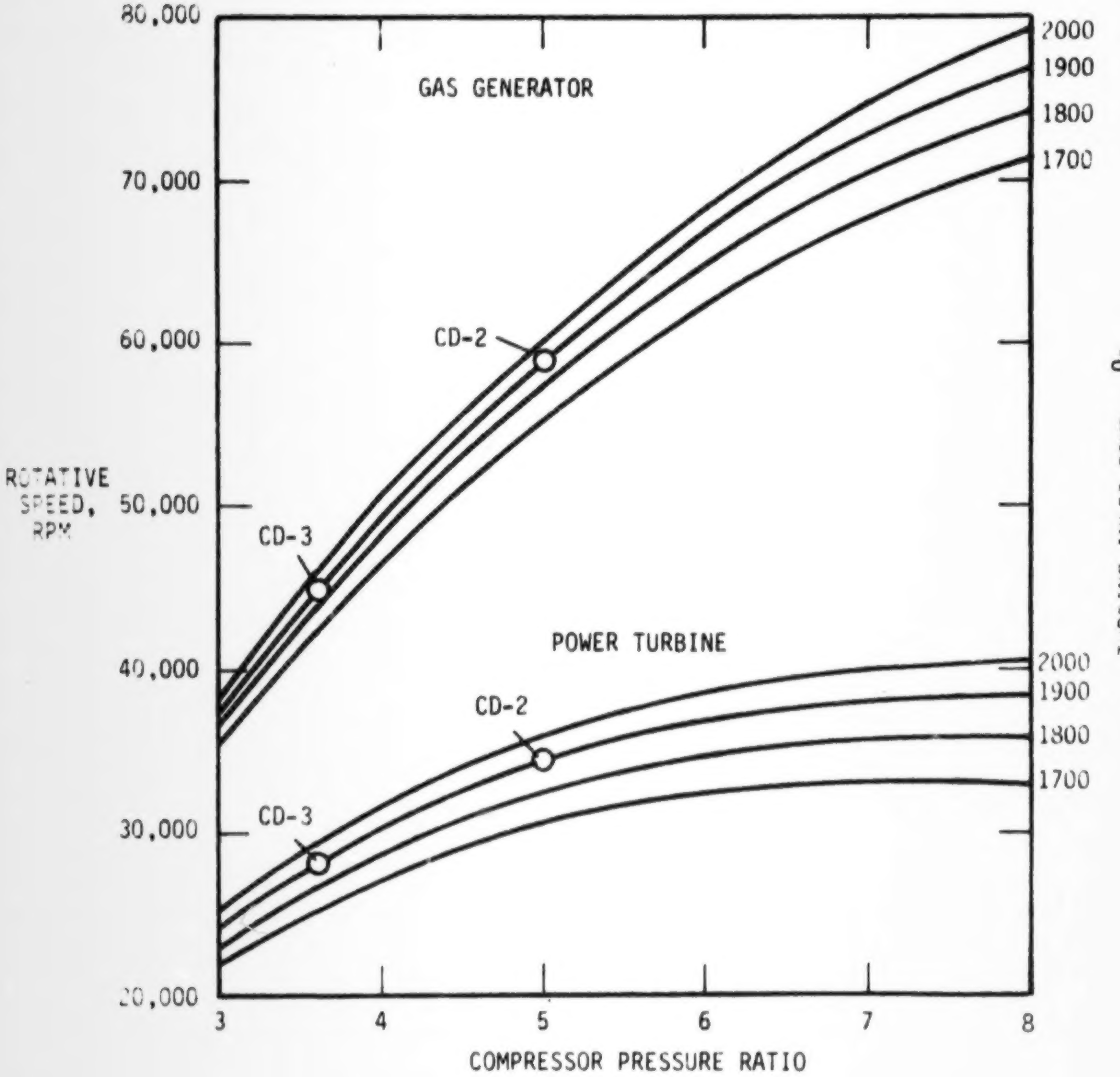


Figure 16. Free Turbine Engine Rotative Speed.

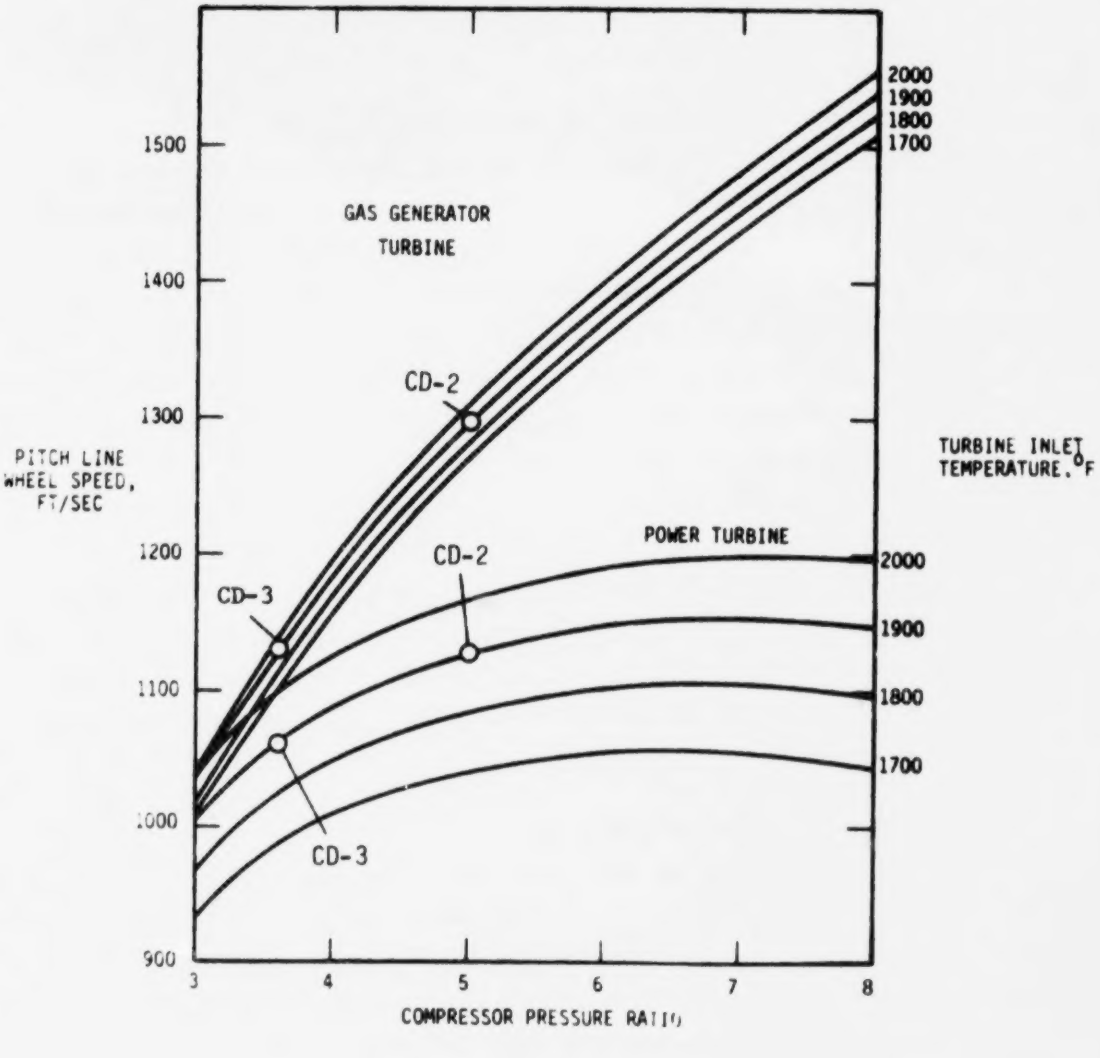


Figure 17. Free Turbine Engine Turbine Wheel Speed.

generator and power turbines are shown in Figures 18 and 19, with the uncooled blade temperatures. These plots indicate the low turbine stresses which are possible by the selection of lower engine pressure ratios, and are used to select the materials required in conjunction with the engine life requirements. The blade temperatures were calculated using the pitchline velocity diagrams.

Points were calculated for six pressure ratios and four turbine inlet temperatures using the inputs for the single shaft engine shown in Table 4. It should be noted that the compressor efficiencies and specific speeds are lower than in Table 3 for the free turbine engine. The reason is that initial calculations indicated high root centrifugal stress in the turbine, which is a function of rotative speed and annulus area. Rotative speed is set by the specific speed of the compressor, and tends to be high for good efficiency. Annulus area is set by the exit Mach number of the turbine, and should be low (around 0.25) to duct the flow into the regenerator with reasonable pressure drop. Therefore, the turbine stress tends to limit the compressor pressure ratio more than for the other configuration. For this reason the compressor specific speed was lowered from the optimum value of about 70, and a penalty is compressor efficiency was accepted. A small penalty in turbine efficiency was taken to raise the exit Mach number to 0.3. The other inputs are similar to those for the free turbine engine.

The compressor tip speed shown in Figure 20 is higher than for the free turbine engine because of the lower efficiency assumption. The rotative speed shown in Figure 21 is lower than for the free turbine engine due to the assumption of lower specific speed. After completion of the design point study, a single shaft engine with variable geometry, CD-1, was selected with a pressure ratio of 3.6. At this low pressure ratio, it was determined that the low specific speed and the corresponding efficiency penalty were too pessimistic and subsequent designs were made for higher specific speed, around 65. The turbine wheel speeds are shown in Figure 22, and are based on the assumption that a two stage turbine was required. Later selection of a low pressure ratio engine, CD-1, resulted in a one stage turbine design.

In all of these design point calculations the compressor and turbine dimensions are different for each point and are determined by wheel speed

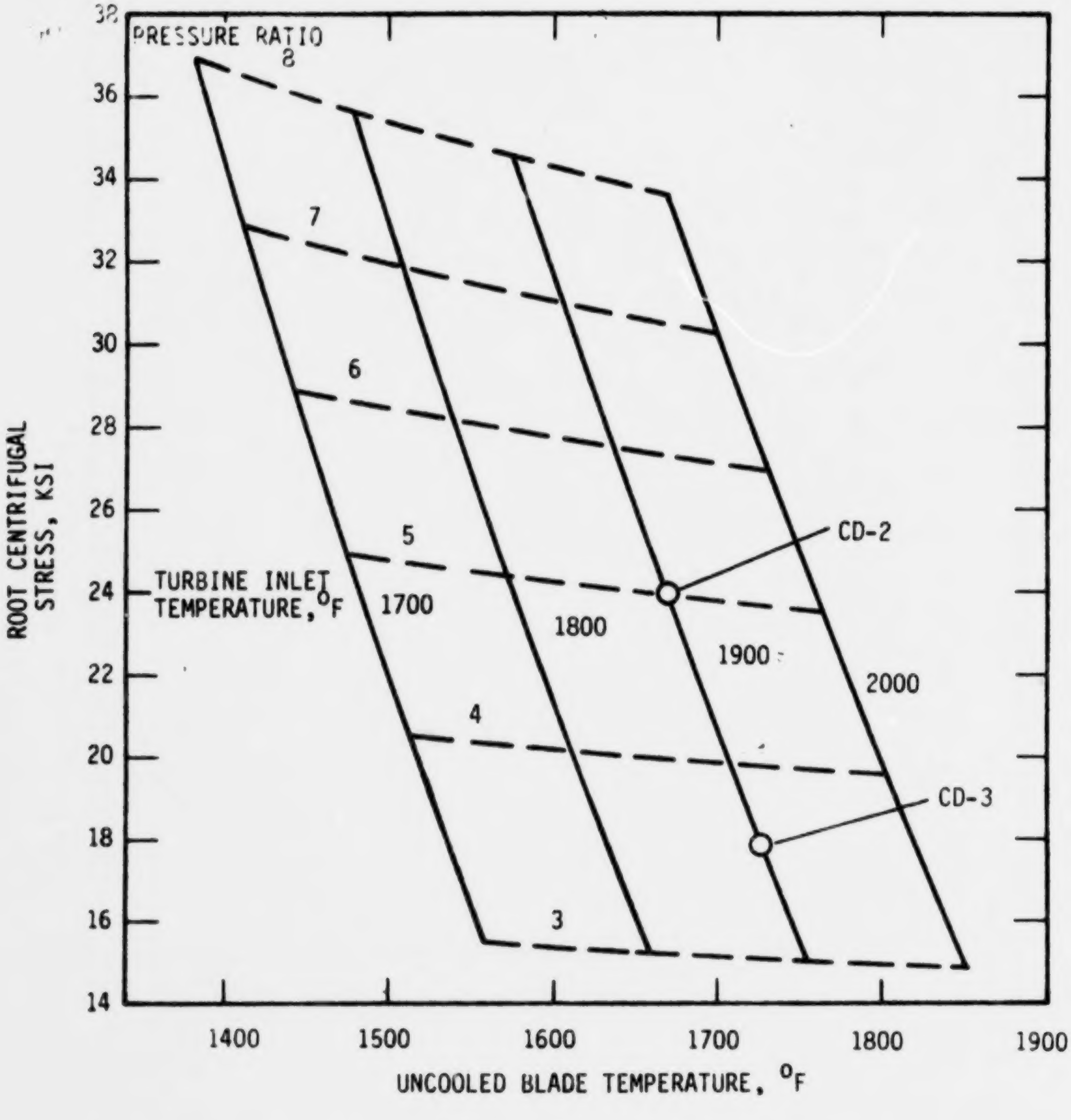


Figure 18. Free Turbine Engine Gas Generator Turbine Stress.

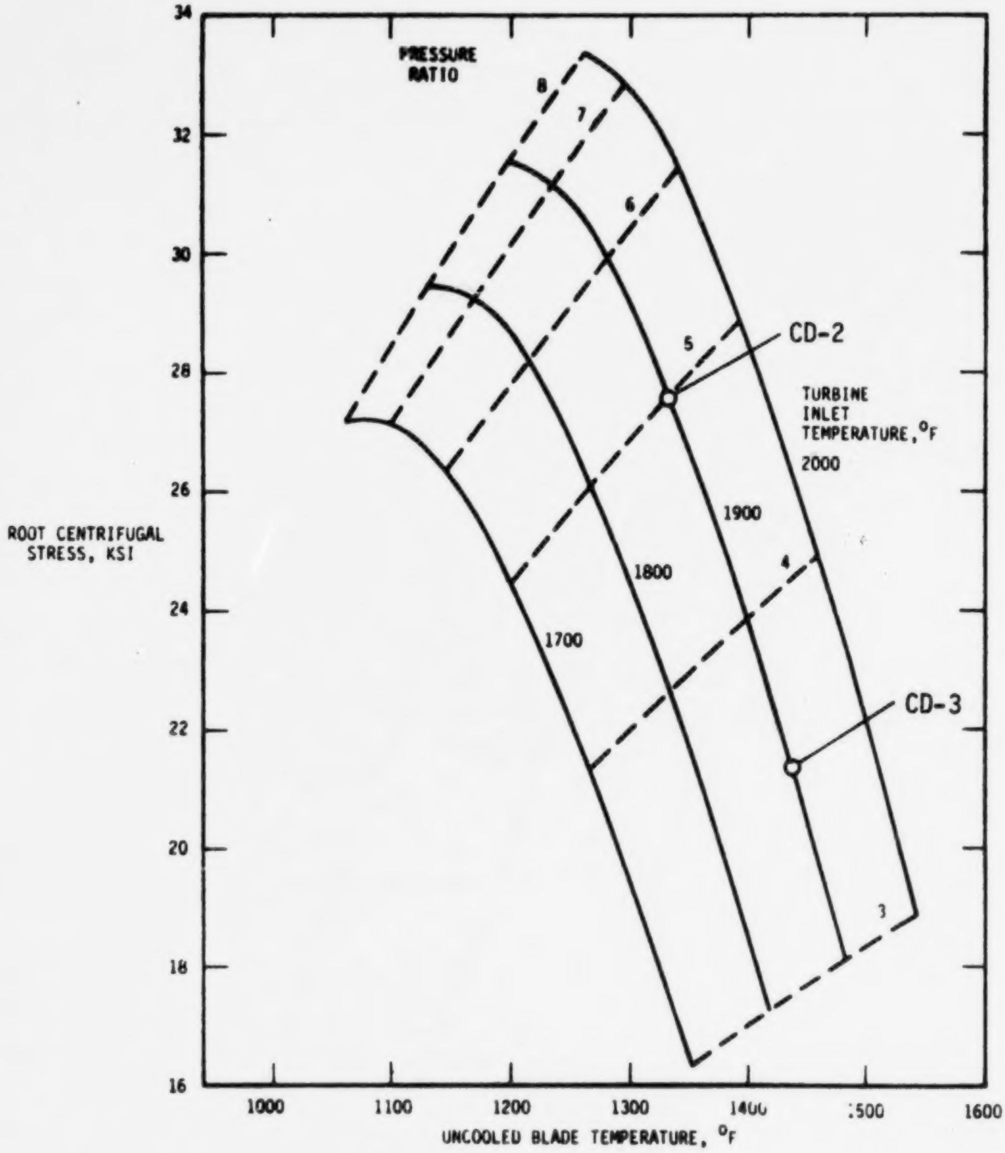


Figure 19. Free Turbine Engine Power Turbine Stress.

Table 4

Single Shaft Engine

Input Variables:

Compressor pressure ratio	3	4	5	6	7	8
Compressor efficiency	.785	.761	.7395	.718	.6945	.67
Turbine efficiency		.84				
Turbine inlet temperature		1700,	1800,	1900,	2000°F	
Cooling flow, %		1	3	5	7	
Compressor inlet temperature		85°F				
Regenerator effectiveness		.9				
Regenerator pressure drop		1% cold side, 3.5% hot side				
Combustor pressure drop		3%				
Tailpipe pressure drop		1%				
Inlet pressure drop		1%				

Assumptions:

Compressor specific speed 52.5, 50, 47.5, 45, 42.5, 40

Turbine exit Mach No. 0.3

Turbine loading, $g \frac{J\Delta h}{U_p^2} = 1.5$

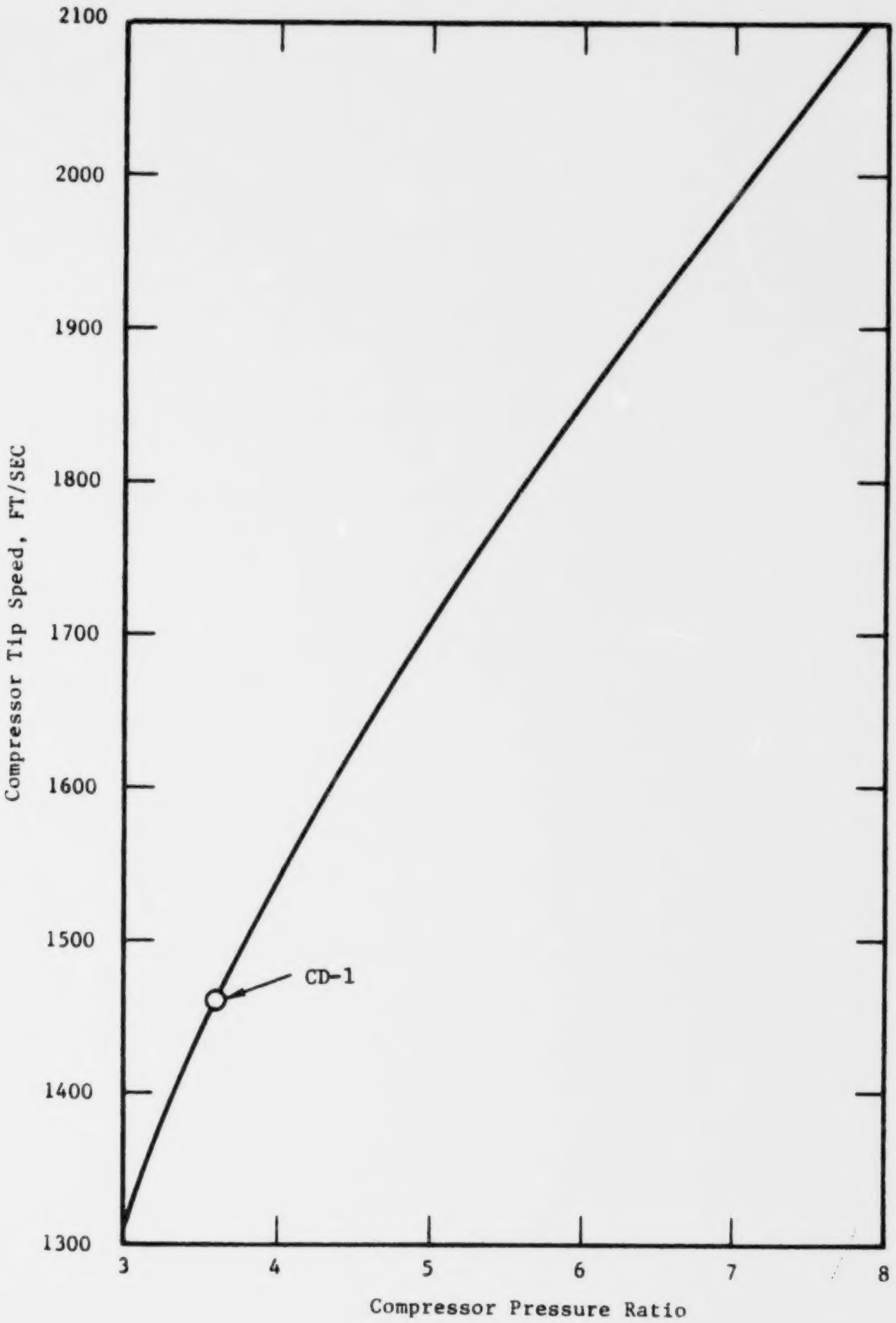


Figure 20. Single Shaft Engine Compressor Tip Speed.

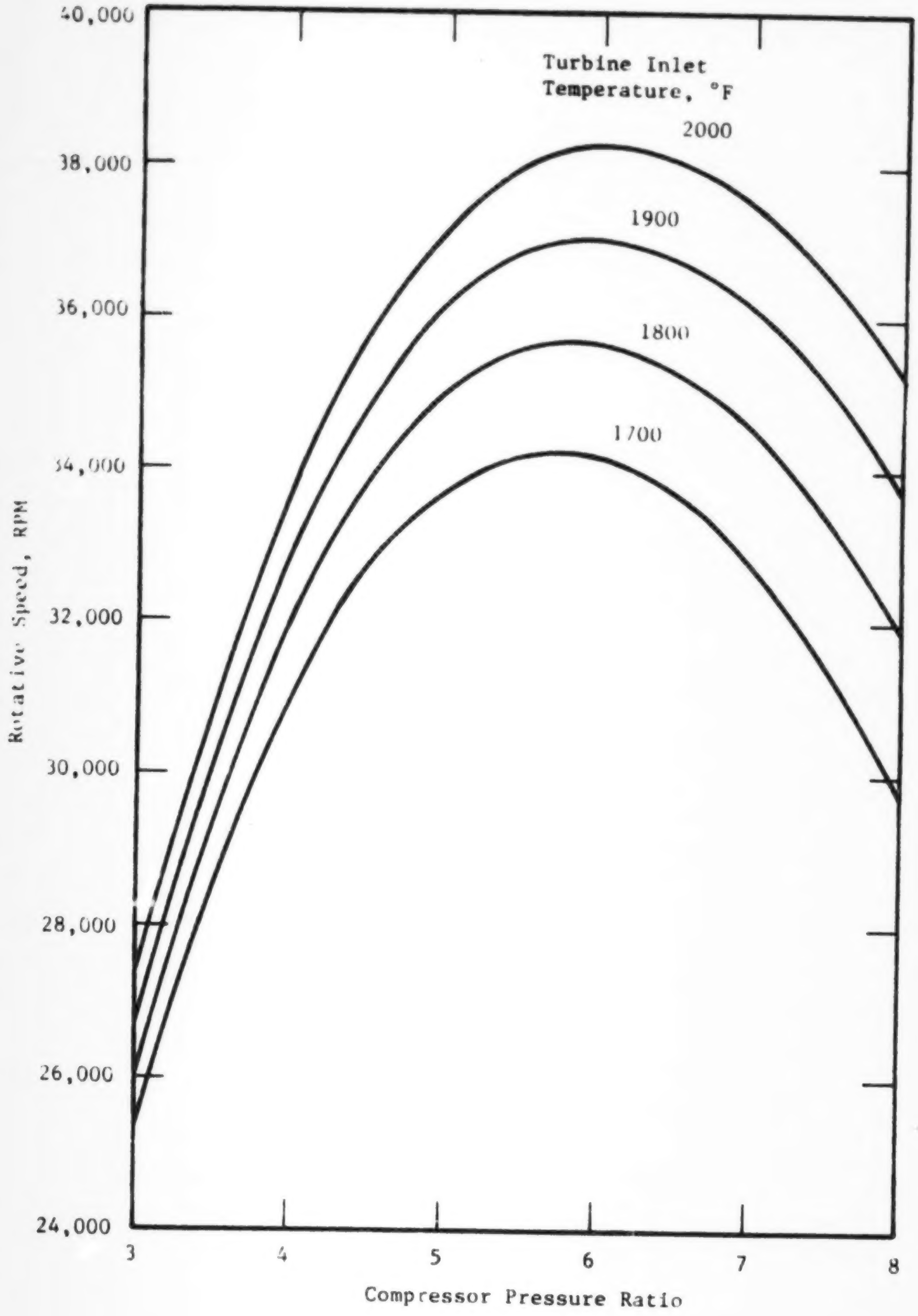


Figure 21. Single Shaft Engine Rotative Speed.

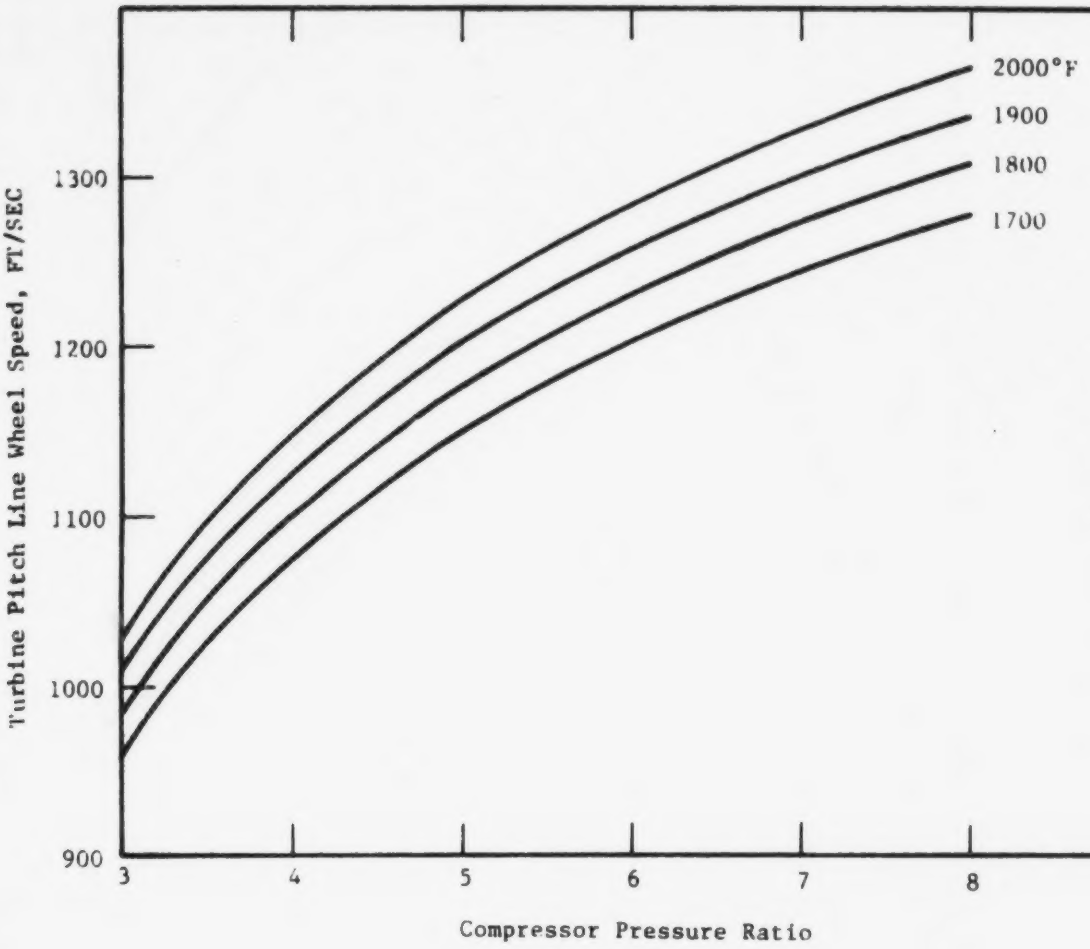


Figure 22. Single Shaft Engine Turbine Wheel Speed.

requirements and rotative speed. The compressor wheel speed is determined by pressure ratio and efficiency, rotative speed is a function of the compressor specific speed, turbine wheel speed is determined by the loading assumption and the specific work requirement.

3.3 Preliminary Selection

In Table 5 are indicated ten engine configurations from which three have been selected for conceptual design and for further analysis and optimization. As shown by Table 5, the ten candidate engines fall into three general types, single shaft engines, free turbine engines, and two spool engines. Criteria which were applied to the selection of the three most promising engines from the ten assigned configurations include the following:

1. Fuel consumption at low loads. (Configurations not meeting the guideline specifications of 8 to 12 lb/hr at idle and 16 to 18 lb/hr at 20% power were automatically eliminated).
2. Estimated relative initial cost as determined by complexity and material requirements.
3. Estimated relative reliability as determined by complexity, and general mechanical design criteria.

3.3.1 Comparative Performance Evaluation

The procedure followed in comparing the various engines was to select cycle parameters and to construct compressor and turbine performance maps for representative engines of the specified configurations to establish full load (design point) performance and then to employ a computer program which carried out off design performance analysis. A summary of the performance comparison of candidate engines is presented in Figure 23, in which engine specific fuel consumption is plotted against engine power output. The following three (regenerative or recuperated) configurations from Table 5 are not represented in Figure 23 for the stated reasons:

Table 5

Optimum Cycle Selection Configurations

Engine Types Modifiers	Single-Shaft W/Transmission	Free Turbine	Two-Spool W/Transmission
Unmodified	No	No	1. Yes
Regenerator	2. Yes	No	No
Recuperator	No	No	3. Yes
Regenerator Var. Pow. Trbn	No	4. Yes	No
Regenerator Var. GG Trbn	No	No	No
Regenerator Var. Comp.	5. Yes	No	No
Regenerator Var. GG Trbn Var. Comp.	6. Yes	No	No
Regenerator Var. Pow. Trbn Var. Comp.	No	7. Yes	No
Regenerator Var. GG Trbn Var. Pow. Trbn	No	No	No
Regenerator Var. Comp. Var. GG Trbn Var. Pow. Trbn	No	8. Yes	No
Power Transfer Recuperator Var. Comp.	No	No	9. Yes
Power Transfer Regenerator Var. Comp.	No	10. Yes	No

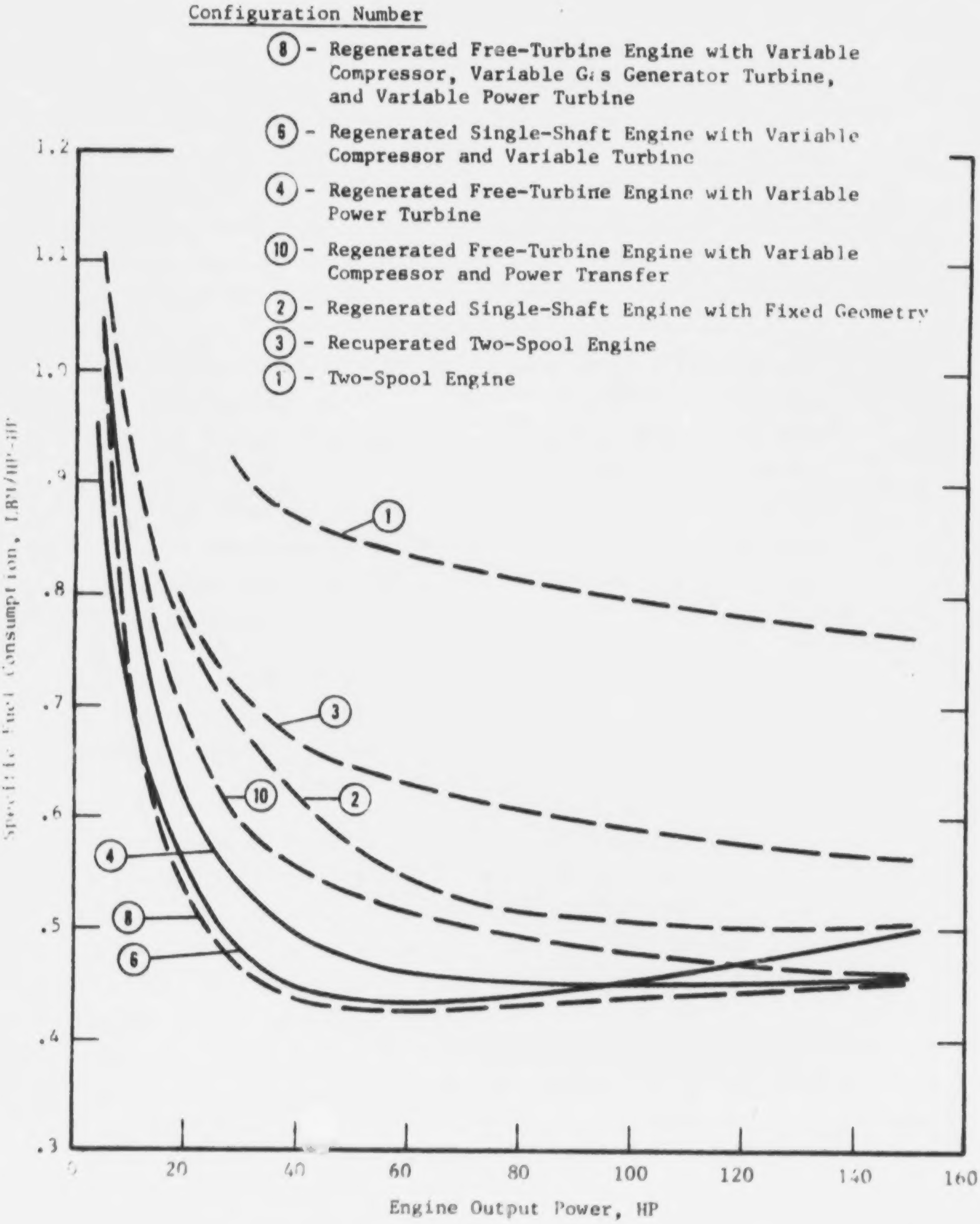


Figure 23. Engine Configuration Performance Comparison.

1. Configuration 5, (single-shaft engine with variable compressor and fixed turbine). No part-load performance advantage (over fixed geometry) was identified for the use of a variable compressor with a fixed turbine.
2. Configuration 7, (free turbine engine with variable compressor and variable power turbine). Only a very marginal advantage could be identified for the addition of variable compressor geometry to a free turbine engine with variable power turbine. This advantage consisted in the capability for a higher stall-limited turbine inlet temperature at low loads. It is believed, however that a turbine inlet temperature of 1700°F can be maintained over the entire power range in a free turbine engine with only a variable power turbine by means of refinements in compressor design and by optimizing the match between compressor and turbine.
3. Configuration 9, (two spool engine with recuperator, variable compressor, and power transfer). No advantages could be identified for the addition of variable compressor geometry and power transfer to the two spool engine with recuperator.

It should be noted that in the case of configuration 10, no advantage was found in the use of variable compressor geometry, that the entire improvement over simple fixed geometry results from power transfer only.

The highest specific fuel consumption was obtained for the two spool, simple cycle (configuration number 1) which had no heat exchanger but had a moderately high pressure ratio (12:1) at the design point. A substantially higher pressure ratio and turbine inlet temperature is required to make this cycle competitive because of the absence of a heat exchanger. The addition of a recuperator with an effectiveness of 0.7 to the two spool engine (configuration number 3) improves the performance and permits the use of the moderate pressure ratio of 6:1. However, the regenerative single shaft engine (pressure ratio < 4.0:1) with fixed geometry (effectiveness 0.85) (configuration number 2) has a lower specific fuel consumption than configuration number 3 because a regenerator can, in general, be made more effective than a recuperator. By using both a variable compressor and turbine on the regenerative single shaft engine (configuration number 6), the specific fuel consumption can be greatly improved especially in the low power range (4 to 60 HP). A regenerative, free turbine engine with power transfer (configuration number 10) has improved performance when compared to the regenerative fixed-geometry single shaft engine (configuration number 2). The free-turbine engine performance is improved markedly when power transfer is replaced by a variable power turbine (configuration number 4). By using variable geometry on the compressor and both turbines (configuration number 8), the free turbine has slightly better fuel consumption than the single shaft engine with fully variable geometry (configuration number 6). Thus, the engines with the three lowest fuel consumption characteristics in order of excellence are configuration numbers 4, 6 and 8. However, the latter three configurations rank 6, 4 and 8 in order of increasing complexity.

Conclusions which were reached upon completion of the preliminary performance comparison of the 10 assigned configurations are the following:

1. The two spool configurations are more complex and costly than either the single shaft or free turbine engines. Furthermore, since they do not appear to be competitive in performance, they should be eliminated from further consideration.
2. The use of power transfer in the free turbine engine does not provide as much light load performance improvement as does the use of variable power turbine geometry, or the use of variable geometry in both turbines as well as in the compressor. The reasons for this appear to be that at part load the power turbine is operating at low efficiency, and that considerable power is dissipated in the power transfer clutch.
3. The three configurations which are best in performance over the entire load range are:

    a. Free turbine engine with variable geometry in both turbines and in the compressor (configuration number 8).
    b. Single shaft engine with variable compressor and variable turbine (configuration number 6).

c. Free turbine engine with fixed compressor and gas generator turbine, and variable power turbine (configuration number 4).

Configurations 6 and 4 above are quite competitive with the alternatives with respect to simplicity, reliability and cost. Configuration 8 has been selected because of its outstanding performance. However, the complexity of this engine is recognized as a distinct disadvantage.

For the stated reasons, the three configurations (4, 6, and 8 of the last paragraph) have been selected. Salient features of these engines are summarized in Tables 6, 7 and 8.

3.3.2 Pressure Ratio Selection

In addition to the selection of three basic engine configurations, it was also necessary to determine optimum pressure ratios for these engines and pursuant to this the following guidelines were established:

1. The efficiency of the regenerative cycle peaks at a pressure ratio of approximately 3.5, (varying slightly with other cycle parameters) and remains high down to a pressure ratio of 2, (assuming cycle temperature ratio is maintained), below which it drops rather rapidly. Therefore, it is desirable to maintain operation at pressure ratios in this range for as much of the engine duty cycle as possible. With fixed geometry engines, this can be accomplished under light load conditions only by designing for a full load pressure ratio of the order of 6, since the speed must drop to the neighborhood of 55%, with a corresponding drop in pressure ratio, in order to cover the load range. With variable compressor and turbine geometry, however, light load operation can be attained at speeds no lower than approximately 70%, making full load pressure ratios in the range of 3.5 to 4.0 quite adequate to maintain a low load pressure ratio of 2.
2. Attainable compressor efficiency increases as pressure ratio decreases.

Table 6

Salient Features of Free Turbine Engine with Variable Power Turbine

(Configuration No. 4)

Regenerated Free Turbine, Variable Power Turbine

Pressure Ratio Range, 4 to 6

Shaft Speeds, 60,000 and 35,000 RPM

Advantages	Disadvantages
1. Good Part-Load Fuel Consumption	1. Requires High Speed on Gas Generator
2. Conventional Transmission	2. Requires High Pressure Ratio Compressor
3. Fixed Geometry Compressor with Optimum Diffuser	3. Braking Requires Turbine Nozzle Reversal
4. No Gas Generator Variable Turbine	4. Requires Two Rotors with Bearings and Seals

Table 7

Salient Features of Single Shaft Engine with Variable Compressor and Turbine

(Configuration No. 6)

Regenerated Single Shaft, Fully Variable

Pressure Ratio Range, 3.2 to 4.0

Shaft Speed, 40,000 RPM

Advantages

1. Good Part-Load Fuel Consumption
2. Very Moderate Rotative Speed
3. Low-Pressure Ratio Compressor
4. Good Braking
5. Only One Rotor with Bearings and Seals

Disadvantages

1. Requires Undeveloped Transmission
2. Must Accelerate all Turbomachinery and Accessories Together
3. Requires Variable Geometry on High-Temperature Turbine

Table 8

Salient Features of Free-Turbine Engine with Variable Compressor and Turbines

(Configuration No. 8)

Regenerated Free Turbine, Fully Variable
Pressure Ratio Range, 3.5 to 4.5
Shaft Speeds, 45,000 and 32,000 RPM

Advantages	Disadvantages
1. Very Good Part-Load Fuel Consumption	1. Complicated by Variable Geometry on Even Gas Generator Turbine
2. Low Pressure Ratio Compressor	2. Braking Requires Turbine Nozzle Reversal
3. Conventional Transmission	3. Requires Two Rotors with Bearings and Seals
4. Moderate Rotative Speeds	
5. Small Change in Gas Generator Speed for Acceleration	

3. The workability of variable compressor geometry is much greater at low pressure ratios (below 3) where inducer and diffuser inlet relative Mach numbers are below 1, than at higher pressure ratios. Workability is the ability to maintain high efficiency and constant pressure ratio over a wide flow range at constant speed, as diffuser vanes are varied. Reference 4 demonstrates the capability of a compressor designed for a low pressure ratio (2.4) to maintain constant efficiency and pressure ratio over a wide flow range as diffuser vane angle is varied at constant speed. Reference 5 indicates that a wide flow range at constant speed may be achieved at higher pressure ratios (4 to 8) but that significant drops in pressure ratio and efficiency are involved.
4. Turbine rotor stress and shaft speed increase with pressure ratio for constant values of compressor design specific speed. This strongly implies that high pressure ratio engines will require costly and strategic materials and that bearing design will be critical. Figure 24 shows the variation of shaft speed and turbine stress with design pressure ratio. In order to use low cost non-strategic turbine materials and also to keep the bearings reliable and inexpensive, low stresses and speeds are necessary.

Performance was calculated for the regenerative free-turbine engine with variable geometry in the power turbine only (configuration No. 4) for design pressure ratios of 4, 5 and 6. The variables were turbine vane setting and the power - engine speed combination determined by a road resistance calculation. It was determined that the lowest fuel consumption was obtained at the maximum turbine inlet temperatures. Although the match-point temperature was 1900°F, that temperature is for accelerations only, and the engine maximum continuous temperature is 1700°F.

As engine speed is reduced to reduce engine output power, the turbine inlet temperature drops and specific fuel consumption increases for a given nozzle vane setting. Higher temperatures and lower fuel consumption are obtained by closing the turbine nozzle vanes. For this study, turbine inlet temperature was held at 1700°F and the turbine vanes

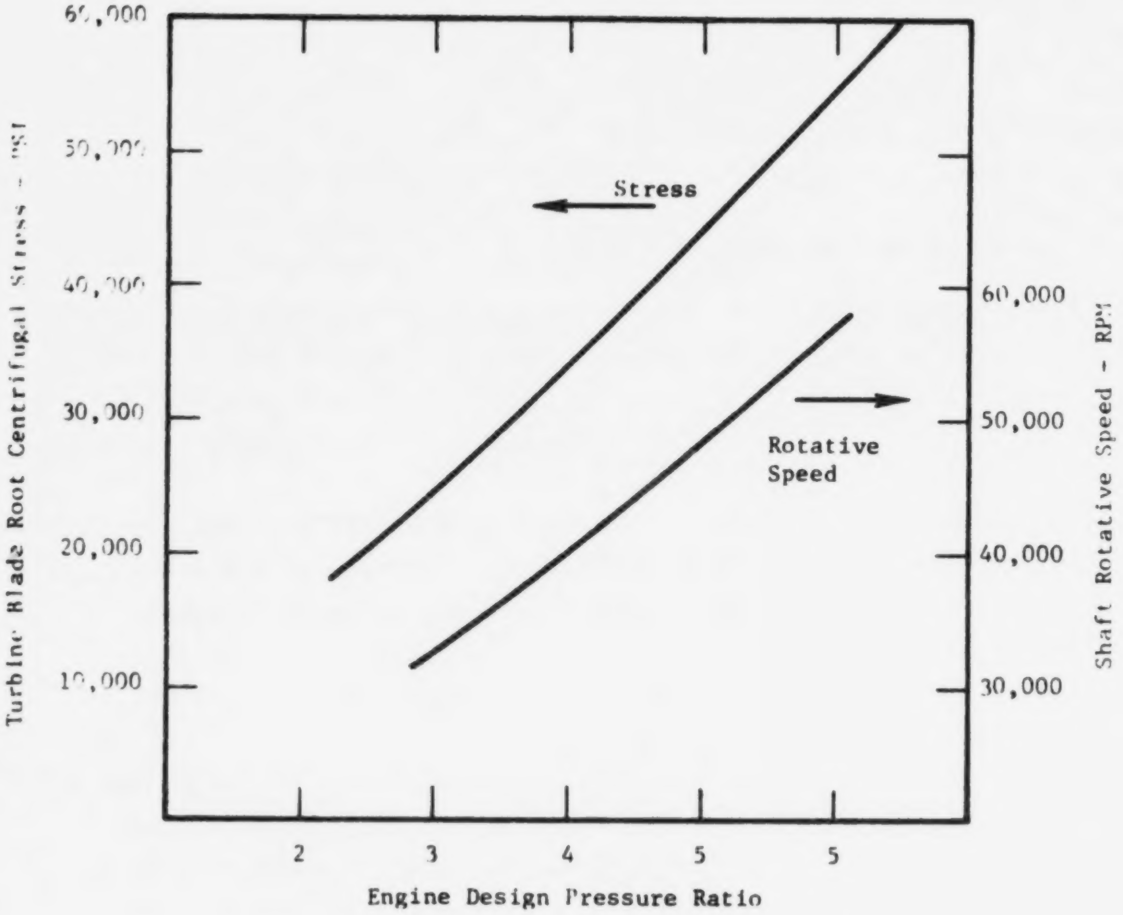


Figure 24. Turbine Rotor Rotative Speed and Stress Versus Design Pressure Ratio (Single Shaft Engine).

were varied to achieve this temperature. The only restriction was compressor stall margin. The fuel economy calculations were based on the expected speed characteristic of the automobile over its lifetime and total fuel costs were calculated. This mission analysis was carried out for the Uniform Simplified Engine Duty Cycle.

		Avg. Speed	Hours	% of Time
1)	Federal Driving Cycle	19.84	1750	50
2)	Simplified Suburban Route (equal times at constant 20, 30 and 40 mph speeds)	30.00	1150	33
3)	Simplified Country Route (equal times at constant 50, 60 and 70 mph speeds)	60.00	600	17
	Totals	30	3500	100

The results are summarized below:

Pressure Ratio	Fuel Mileage, mpg	Fuel Cost
4	16.61	$ 1964.
5	17.68	1845.
6	18.02	1810.

The breakdown of the total fuel cost into the parts of the driving cycle is shown in Figure 25. Although this preliminary mission analysis indicated that the fuel consumption was less for the design pressure ratio of 6, a pressure ratio of 5 was selected because its total fuel cost for the life of the automobile (105,200 miles) was only $35 more (out of approximately $1,800) than for a pressure ratio of 6. This judgement was based upon the probability that the engine with a pressure ratio of 6 would cost at least $35 more than an engine with a pressure ratio of 5. The selected engine (configuration No. 4 with a pressure ratio of 5) was redesignated CD-2. See Table 2 for engine design parameters.

Off-design performance was also calculated for the regenerative single-shaft engine with variable compressor and turbine geometry (configuration No. 6) at design pressure ratios of 3.2, 3.6, and 4.0. The minimum SFC for a given power is sought for each design pressure ratio by varying engine speed and compressor and turbine stator settings. A preliminary mission analysis was also made for the single shaft engine and the results are summarized below:

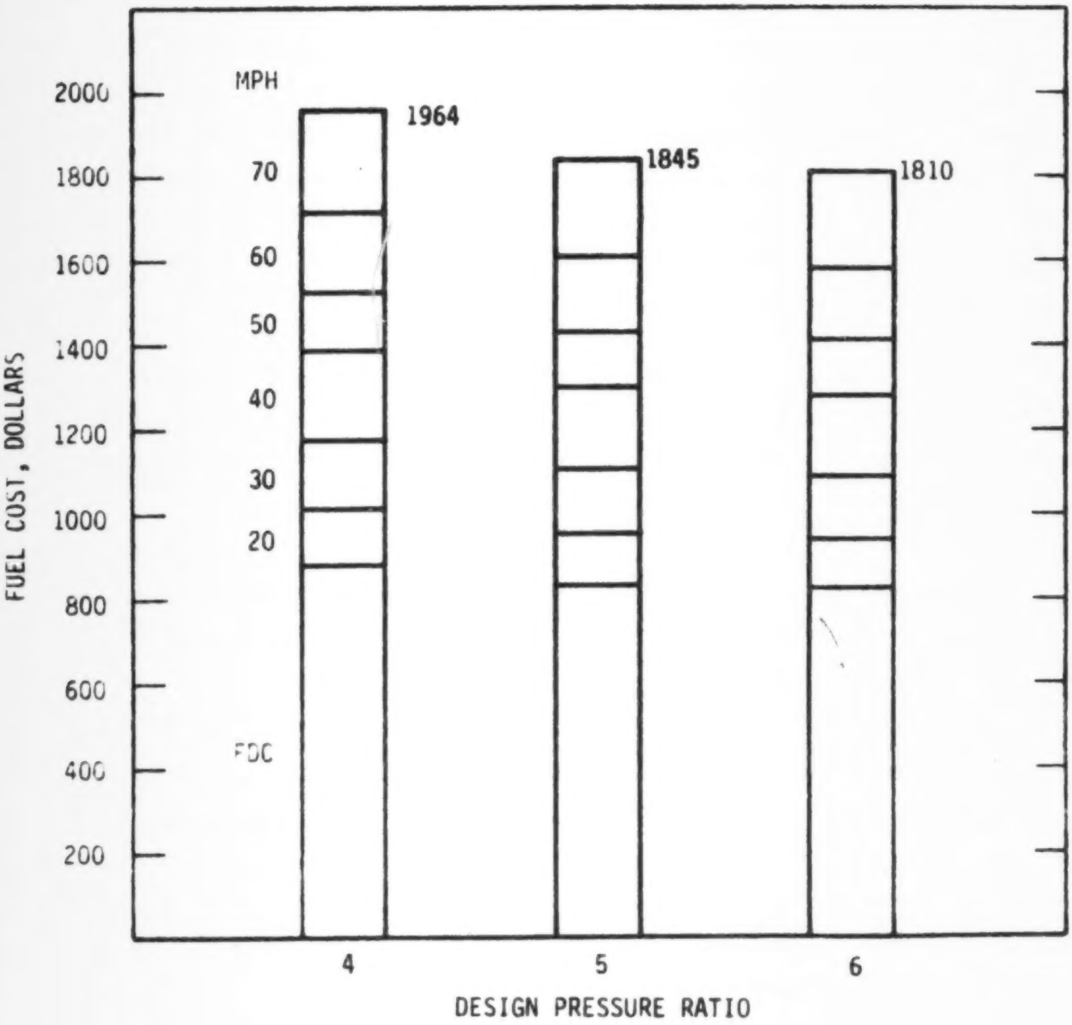


Figure 25. Free Turbine Engine, Variable Power Turbine Total Mission Fuel Cost.

Pressure Ratio	Fuel Mileage, mpg	Fuel Cost
3.2	17.27	$ 1888.
3.6	17.54	1860.
4.0	17.67	1846.

A breakdown of the total fuel cost is shown in Figure 26. Although the highest pressure ratio engine has the best fuel consumption, the pressure ratio of 3.6 was selected since its total fuel cost is only $14 more than the higher pressure ratio and the compressor variable geometry is more effective at the lower pressure ratios. The selected engine (configuration No. 6 with a pressure ratio of 3.6) was redesignated CD-1. See Table 2 for engine design parameters.

The pressure ratio for the regenerative free-turbine engine with variable geometry on the compressor and both turbines was not determined. This engine was later dropped from consideration because the improvement in performance was too small for the added complexity.

The engines designated CD-1 and CD-2 were carried through conceptual design. The salient features of these two engines are summarized in Table 9.

3.4 Design

Design effort was accomplished at two levels. The conceptual design effort resulted in layout drawings from which weight and costs could be estimated. The preliminary design effort was in greater depth and took advantage of weight and cost-saving innovations stemming from the evaluation of the conceptual designs.

3.4.1 Conceptual Design

Conceptual designs were carried out on the CD-1 and CD-2 engines identified in the previous section for further study.

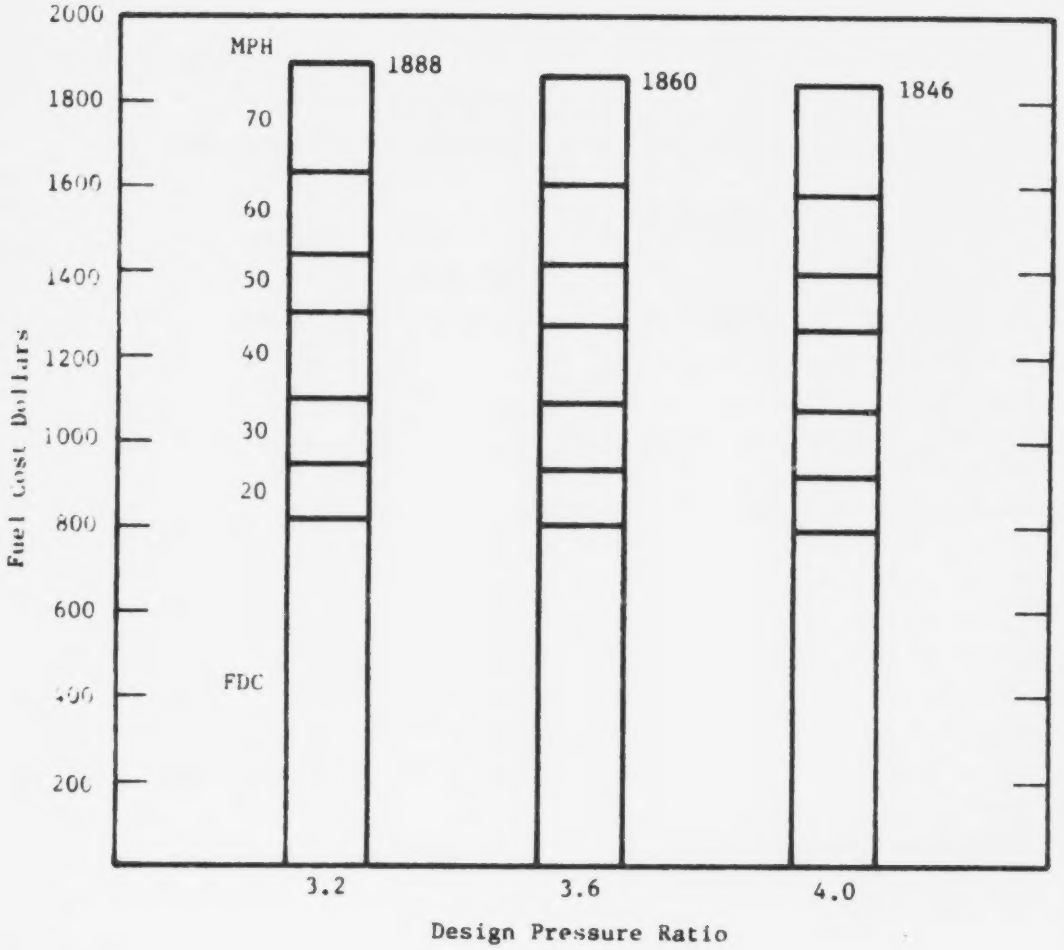


Figure 26. Variable Geometry Single Shaft Engine Total Mission Fuel Cost.

Table 9

Conceptual Design Engines

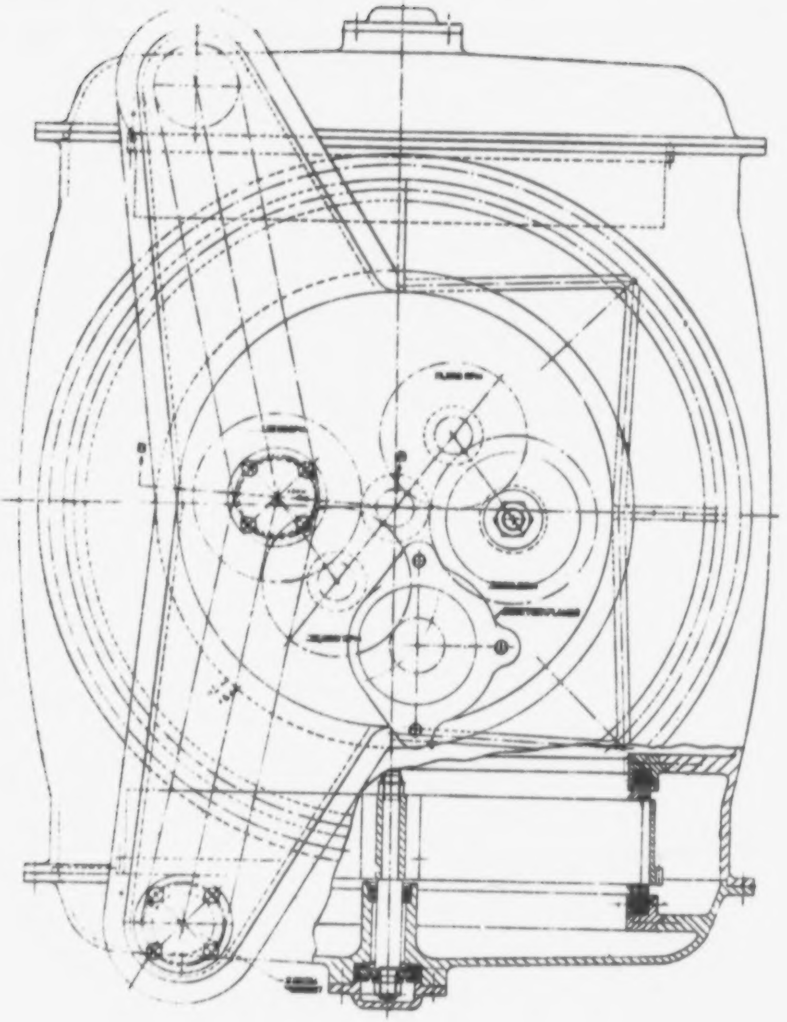
	CD-1	CD-2
Engine type	Single shaft	Free turbine
Heat exchanger	Regenerator	Regenerator
Variable compressor	Yes	No
Variable compressor turbine	Yes	No
Variable power turbine	Does not apply	Yes
Compressor speed, rpm (max.)	40,000	55,000
Power turbine speed, rpm (max.)		35,000
Pressure ratio	3.6	5
Turbine inlet temperature, °F (max) (for accelerations)	1900	1900
Turbine inlet temperature, °F (normal continuous)	1700	1700
Idle fuel flow, lb/hr	8.0	5.5
Max. fuel flow, lb/hr (normal continuous)	65	58
Special combustor provision	None	None
Accessory drive location	Compressor	Gas Generator Compressor

3.4.1.1 Free-Turbine Engine with Variable Power Turbine (CD-2)

Shown in Figure 27 is the CD-2 free-turbine engine with variable power turbine. The materials list which is keyed to the drawing is presented in Table 10.

Description - Air enters the engine inlet through the filter which extends in a 180° arc over the top side of the engine, passes through the centrifugal compressor inlet, through the compressor impeller, the pipe diffuser, and to the forward half of the two regenerator discs. The flow then proceeds through the forward halves of the regenerators into the forward housing space enclosing the gas generator turbine inlet scroll and the combustor liner which is assembled to the inlet end of this scroll. Flow enters the turbine scroll through the combustor and passes to the turbine nozzle which is welded to the aft opening of the scroll. From the gas generator turbine, flow passes through a transition passage to the power turbine. The power turbine nozzle vanes are variable. From the power turbine discharge diffuser, the flow exhausts through the rear halves of the regenerator discs to the exhaust ducts attached to the regenerator covers.

The engine components are structurally integrated by the main housing, which is a nodular iron casting. The housing has two forward end flanges. On one of the flanges, the compressor front shroud, air filter, and accessory gear case are mounted, while the compressor rear shroud and the attached gas generator assembly are mounted on the other flange. At the sides of the main housing are flanges for attachment of the regenerator seals and the regenerator covers which support the regenerator rotor discs and the regenerator drive gears. On the top of the housing is a mounting flange for the combustor cover to which are attached the fuel inlet pipe and the spark plug. An important part of the housing is the central bulkhead into which the gas generator and power turbine assemblies are sealed by means of piston ring seals. At the rear of the housing are mounting flanges for the power-turbine reduction-gear assembly and the transmission. Inside surfaces of the main housing are insulated against hot gas.



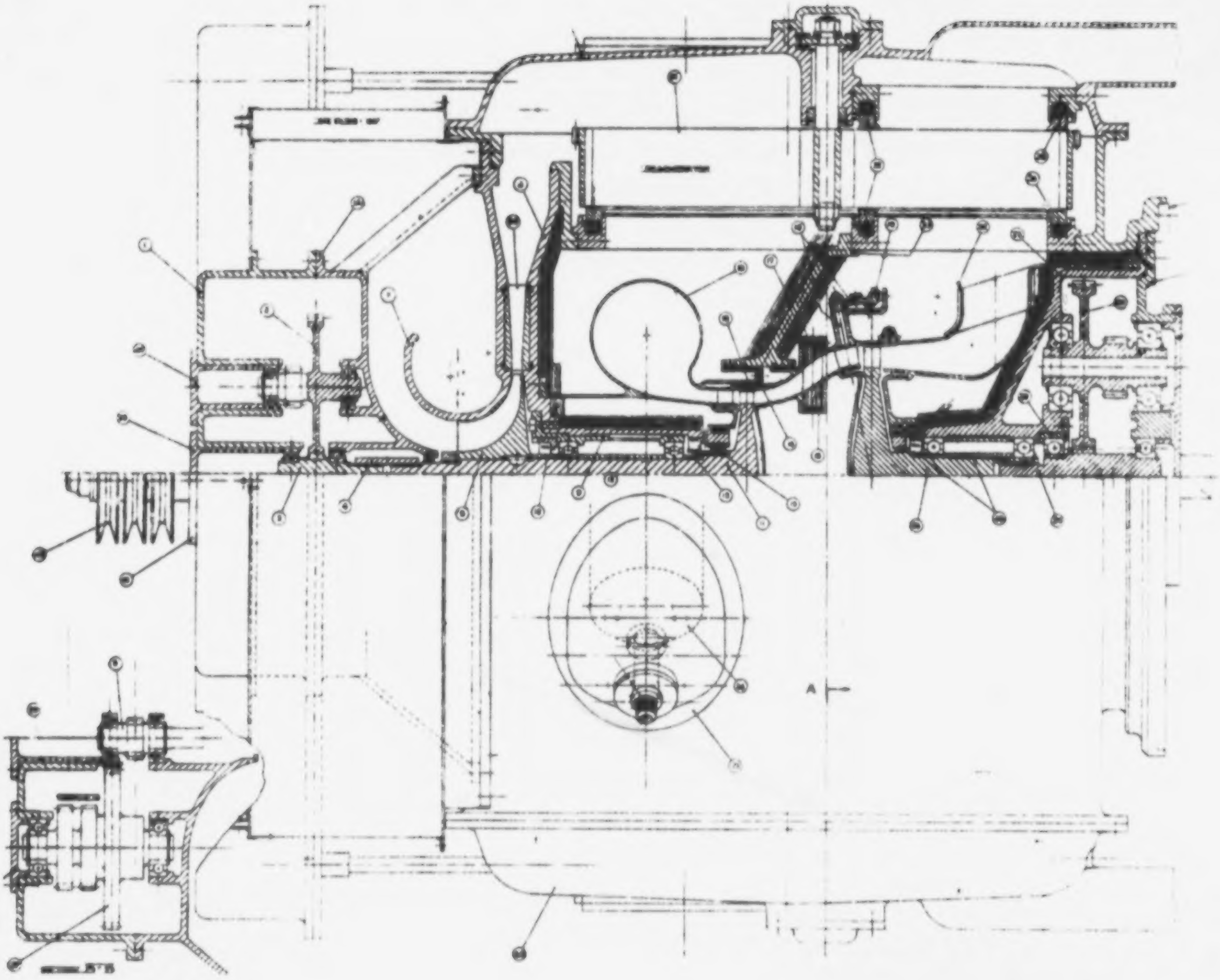


Figure 27. Regenerated Free Turbine Engine, Power Turbine Variable (CD-2)

Table 10

Materials List

Regenerated Free Turbine Engine With Variable Power Turbine (CD-2)

1.	Accessory Drive Case (outer)	Nodular Iron
1A.	Accessory Drive Case (inner)	Nodular Iron
2.	Accessory Power Take-off Gear	8620 Steel
3.	Accessory Power Take-off Shaft	4340 Steel
4.	Splined Coupling	4340 Steel
5.	Compressor Rotor	Aluminum
6.	Regenerator Cover	Nodular Iron
7.	Compressor Front Shroud	Nodular Iron
8.	Compressor Rear Shroud	Nodular Iron
9.	Bearing Mount & Oil Dampner	4340 Steel
10.	Gas Generator Turbine Scroll Shroud	MAR-M-509
11.	Turbine Rotor	IN 738
12.	Spacers	4340 Steel*
13.	Labyrinth Seal	4340 Steel
14.	Turbine Shroud Support	IN 738*
15.	Gas Generator Tip Seal	IN 738*
16.	Turbine Transition Inner Body	304 SS
17.	Turbine Transition Mounting Pins	IN 738*
18.	Power Turbine Nozzle	
	A) Partitions	N-155
	B) Portion Drill Gear	304 SS
19.	Power Turbine Actuator Ring	304 SS
20.	Turbine Diffusers	Hastelloy X
21.	Combustor Cover	304 SS
22.	Turbine Diffuser	Hastelloy X
23.	Output Gear Case	Nodular Iron
24.	Power Turbine Rotor	CRM 6D
25.	Spacers	4340 Steel
26.	Splined Gear Shaft	4340 Steel
27.	Output Gear Case Cover	Nodular Iron
28.	Bearing Housing	4340 Steel
29.	Output Gear	8620 Steel
30.	Output Gear Case End Plate	Nodular Iron
31.	Regenerator Rotor	Cercor (Corning Glass)
32.	Regenerator Seals	Stellite 3
33.	Main Housing	Nodular Iron
34.	Combustion Liner	Hastelloy X
35.	Regenerator Seal	Stellite 3
36.	Regenerator Seal	Stellite 3
37.	Accessory Drive Gear	8620 Steel

*Preliminary selection; may be changed because of high strategic materials content.

Table 10 (Cont'd.)

Materials List

Regenerated Free Turbine Engine With Variable Power Turbine (CD-2)

38.	Bearing Retainer	Nodular Iron
39.	Bearing Retainer	Nodular Iron
40.	Bearing Retainer	Nodular Iron
41.	Bearing Retainer	Nodular Iron
42.	Sheave - V Belt	Nodular Iron
43.	Bushing	Teflon
44.	Bushing Seal	304 SS
45.	Bushing Holder	304 SS
46.	End Cap	Nodular Iron
47.	Regenerator Shaft	304 SS
48.	Turbine Transition Outer Body	304 SS
49.	Turbine Rotor Tip Wiper	Hastelloy X
50.	Turbine Diffuser Supports	Hastelloy X
51.	Turbine Diffuser	Hastelloy X
52.	Labyrinth Rotor Seal Wiper	Hastelloy Honeycomb
53.	Turbine Rotor Spanner Nut	304 SS
54.	Seal Support	304 SS
55.	Guide Buttons	304 SS
56.	Filter Support Bottom Bracket	1010 Steel
57.	Filter Support Top Bracket	1010 Steel
58.	Filter Support Leg Bracket	1010 Steel
59.	Filter Clamp, Lower	1010 Steel
60.	Filter Clamp, Upper	1010 Steel
61.	Filter Bracket, Lower	1010 Steel
62.	Gas Turbine Cover	304 SS
63.	Bleeder Tube	304 SS
64.	Set Screw	304 SS
65.	Washer	304 SS
66.	Fuel Injection Nozzle	304 SS
67.	Filter Rest	1010 Steel
68.	Worm Gear	8620 Steel
69.	Front Regenerator Drive Housing	Nodular Iron
70.	Rear Regenerator Drive Housing	Nodular Iron
71.	Regenerator Drive (Worm & Shaft)	8620 Steel
72.	Spring Retainer	304 SS
73.	Floating Spring Pad	304 SS
74.	High Temperature Coil Springs	Purchase Item
75.	Spring Retainer	304 SS
76.	Spring Retainer	304 SS
77.	Floating Spring Pad	304 SS
78.	Bushing	Teflon
79.	Regenerator Drive Gear	8620 Steel
80.	Floating Spring Pad	304 SS

Table 10 (Cont'd.)

Materials List

Regenerated Free Turbine Engine With Variable Power Turbine (CD-2)

81.	Sheave Shaft	Shaft Steel
82.	Sheave Steel	8620 Steel
83.	Retaining Nut	4140 Steel
84.	Regenerator Drive Cover	Nodular Iron
85.	Coupling	4140 Steel
86.	Shaft Cover Plate	Nodular Iron

The gas generator assembly includes the following items: compressor impeller, the turbine rotor and shaft, upon which the compressor impeller is mounted, the compressor rear shroud-bearing support sleeve, upon which are mounted two ball bearings (the rear one being resiliently mounted), the turbine scroll-nozzle assembly, and the combustion liner, which is assembled to the inlet end of the scroll through an opening in the main housing. The gas generator portion is assembled as a unit to one of the main housing forward flanges. It is sealed into the forward main housing cavity through piston ring seals. Compressor discharge air is admitted to the space between the bearing support tube and the turbine scroll. From this space, cooling air is passed through a labyrinth seal over the forward face of the turbine disc. Full rotative speed of the gas generator is 60,000 RPM.

The power turbine-reduction gear assembly consists of the power turbine rotor and its bearings, the variable power turbine nozzle and nozzle actuating linkage assembly, the transition duct assembly, the turbine discharge diffuser, and the reduction gear assembly. The reduction gear drive pinion is mounted on its own bearings and is connected by means of a spline to the power turbine shaft. The variable nozzle inner and outer flow annulus walls are spherical so that nozzle vane clearance is unaffected by rotation. Individual vanes are attached to gear segments which mesh with an actuating ring gear. Approximately 120° of vane rotation is available between the low power "closed" position and the reversed "braking" position. The entire power turbine-reduction gear assembly is mounted on a rear flange of the main housing. The assembly is sealed into the main housing aft cavity by means of piston ring seals. Full speed of the power turbine is 35,000 RPM.

The two regenerator assemblies are positioned symmetrically at the sides of the engine and consist of the ceramic honeycomb discs, a central shaft mounted on bearings capable of supporting both radial and thrust loads, a metallic rim carrying a ring gear, and seal assemblies consisting of spring-mounted Stellite 3 rings sliding in the grooves of mounting rings. Partial pressure balancing is employed on these seals. The regenerator discs are driven by pinion gears meshing with the disc rim ring gear. The pinions are driven from the accessory gear case through worm gears mounted in the regenerator cover.

The accessory gear case is mounted from one of the forward flanges on the main housing. The accessory drive pinion is mounted on its own bearings and is connected by a spline to the front end of the gas generator shaft. The accessory drive shaft, carrying sheaves for driving engine-mounted accessories runs at 1/16 gas generator speed. Regenerator drive shafts are driven through chain drives. An electric starter drives the gas generator through one of the reduction gears.

Design Problems - Design problems which were identified in connection with the conceptual design of Figure 27, and which were resolved in Preliminary Design, include the following:

1. Exposure of power turbine nozzle actuating gears and linkage to turbine discharge gas. Although it may be possible to identify gear materials which operate satisfactorily at elevated temperature, it appears more promising to enclose the actuating gears in a compressor discharge air environment. Such an approach also will expose the main housing central partition to compressor discharge air on one side and thus aid in limiting the temperature of this partition to an acceptable level (950°F max).
2. Regenerator seals and drive. As an alternate to the use of spring-loaded seals on both sides of the regenerator discs, it seems more promising to spring load the disc from the outside only, and to allow the disc to bear against single flat sealing surfaces on the inside. This will remove force and moment reactions from the central shaft, except those due to the disc weight.

   It may be more economical to drive the regenerator from the center rather than from the rim.

3.4.1.2 Single-Shaft Engine with Fully Variable Geometry (CD-1)

Shown in Figure 28 is the CD-1 single shaft engine design concept with fully variable turbomachinery. The materials list which is keyed to the drawing is presented in Table 11.

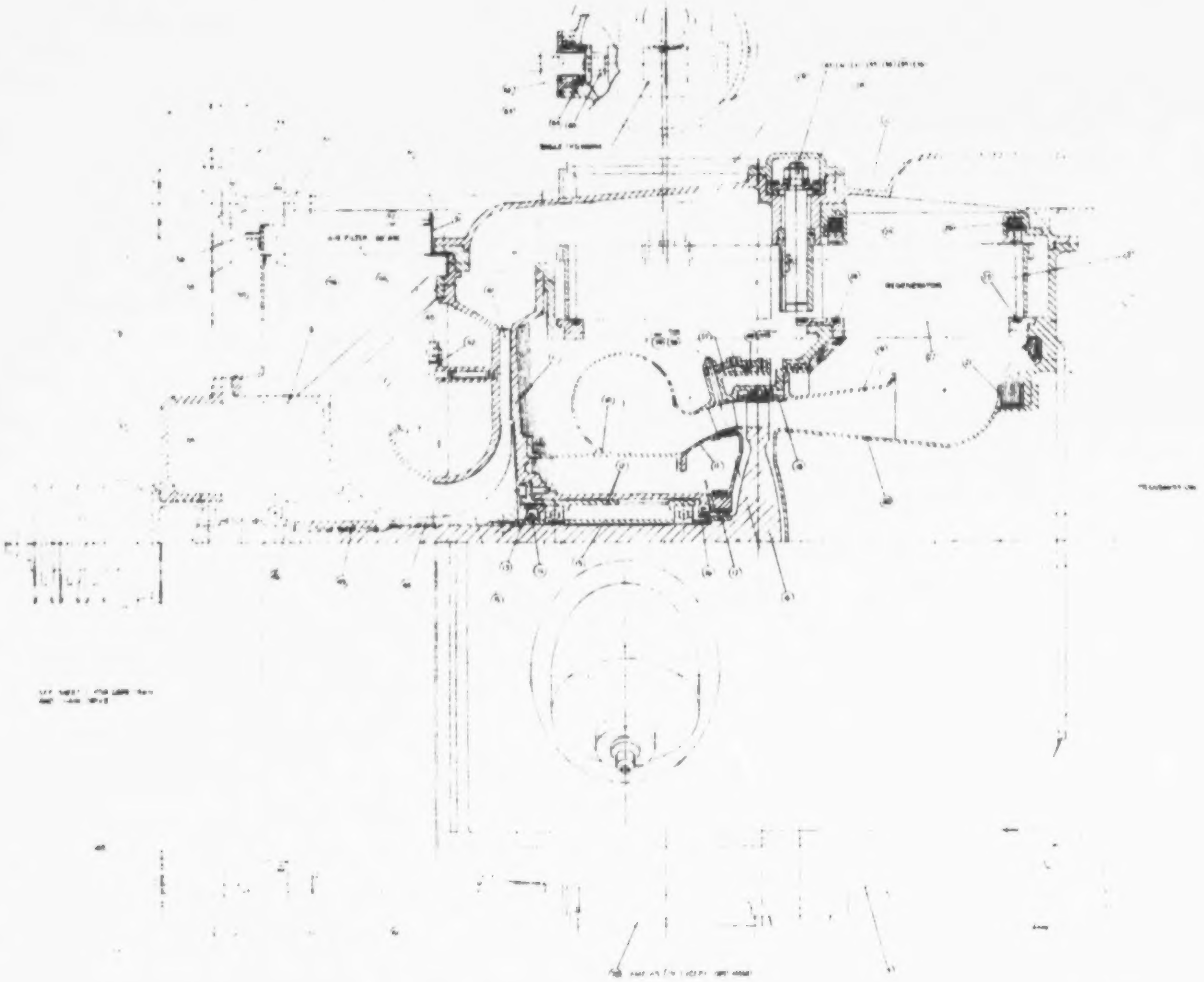

REGENERATOR

Figure 28. Regenerated Single Shaft, Engine, Fully Variable (CD-1)

Reproduced from best available copy.

Table 11

Materials List

Regenerated Single Shaft Engine - Fully Variable Geometry (CD-1)

1.	Main Housing	Nodular Iron
2.	Regenerator Cover, Right Side	Nodular Iron
3.	Regenerator Cover, Left Side	Nodular Iron
4.	Turbine Rotor	IN 738
5.	Compressor Rotor	Aluminum (Cast)
6.	Compressor Rear Shroud	Nodular Iron
7.	Compressor Front Shroud-Outer	Nodular Iron
8.	Compressor Front Shroud-Inner	Nodular Iron
9.	Gear Case	Nodular Iron
10.	Turbine Scroll-Shroud	MAR-M-509
11.	Turbine Stator Inner Shroud	MAR-M-509
12.	Turbine Shaft Bearing Mount	4340 Steel
13.	Turbine Shaft Seal Mount	4340 Steel
14.	Turbine Shaft Spacer	4340 Steel
15.	Turbine Shaft Spacer	4340 Steel
16.	Turbine Shaft Spacer	4340 Steel
17.	Labyrinth Seal	4340 Steel
18.	Turbine Tip Seal	IN 738*
19.	Turbine Diffuser - Outer Shroud	Hastelloy X
20.	Turbine Diffuser - Inner Shroud	Hastelloy X
21.	Insulation Shield	304 SS
22.	Regenerator Seal Support	304 SS
23.	Regenerator Seals	Stellite 3
24.	Regenerator Seals	Stellite 3
25.	Regenerator Seal Support	304 SS
26.	Regenerator Seal Support	304 SS
27.	Regenerator Rotor	Cercor (Corning Glass)
28.	Regenerator Shaft Cover	Nodular Iron
29.	Regenerator Drive Access Cover	Nodular Iron
30.	Regenerator Shaft	304 SS
31.	Regenerator Support - Nut	4340 Steel
32.	Regenerator Support - Washer	4340 Steel
33.	Regenerator Support - Thrust Washer	4340 Steel
34.	Regenerator Support - Thrust Disk	4340 Steel
35.	Regenerator Support - Thrust Bearing	Teflon
36.	Regenerator Support - Radial Bearing	Teflon
37.	Turbine Nozzle Adj. Arm	IN 738*
38.	Turbine Nozzle and Shaft	IN 738*
39.	Turbine Nozzle Lock Nut	304 SS
40.	Turbine Nozzle Actuator Ring	304 SS

*Preliminary selection; may be changed because of the high strategic material content.

Table 11 (Cont'd.)

Materials List

Regenerated Single Shaft Engine - Fully Variable Geometry (CD-1)

41.	Compressor Diffuser Vane	Nodular Iron
42.	Compressor Diffuser Adj. Arm	4340 Steel
43.	Compressor Diffuser Actuator Ring	4340 Steel
44.	Compressor Lock Nut	4340 Steel
45.	Coupling, Splined	4340 Steel
46.	Power Take-off Shaft	4340 Steel
47.	Combustor Cover	304 SS
48.	Burner	304 SS
49.	Combustor Liner	Hastelloy X
50.	Air Filter Bracket	1010 Steel
51.	Air Filter Bracket	1010 Steel
52.	Air Filter Bracket	1010 Steel
53.	Air Filter Bracket	1010 Steel
54.	Air Filter Bracket	1010 Steel
55.	Air Filter (Purchased Item	---
56.	Air Filter Bracket	1010 Steel
57.	Bearing Retainer	Nodular Iron
58.	Regenerator Drive Shaft Gear	8620 Steel
59.	Bearing (Purchased Item)	---
60.	Coupling	4340 Steel
61.	Regenerator Drive Shaft	4340 Steel
62.	Seals (Purchased Item)	---
63.	Cover	Nodular Iron
64.	Bushing	4140 Steel
65.	Coupling	4140 Steel
66.	Bearing (Purchased Item)	---
67.	Bearing Retainer	Nodular Iron
68.	Bearing Retainer	Nodular Iron
69.	Regenerator Drive Shaft Gear	8620 Steel
70.	Regenerator Drive Access Cover	Nodular Iron
71.	Chains (Purchased Item)	---
72.	Bearing Retainer	Nodular Iron
73.	Accessory Drive Gear	8620 Steel (Carburized)
74.	Chain (Purchased Item)	---
75.	Drive Shaft	4340 Steel
76.	Sheave, V Belt	Nodular Iron
77.	Bearing Retainer	Nodular Iron
78.	Washer	304 SS
79.	Nut	4340 Steel
80.	Drive Shaft Extension	4340 Steel
81.	Speed Reduced Casing	Nodular Iron
82.	Speed Reduced Casing	Nodular Iron
83.	Worm Gear	8620 Steel

Table 11 (Cont'd.)

Materials List

Regenerated Single Shaft Engine - Fully Variable Geometry (CD-1)

84.	Worm Gear	8620 Steel
85.	Pinion Gear	8620 Steel
86.	Spacers	8620 Steel
87.	Bushings	Bronze
88.	Seals (Purchased Item)	---
89.	Screws (Purchased Item)	---
90.	Bearing Retainer	Nodular Iron

Description - The structural arrangement is similar to that of Figure 27 except that there is only one turbine. The compressor-turbine assembly is similar to the gas generator assembly of Figure 27. Both compressor diffuser vanes and turbine nozzle vanes are variable. There is no separate power turbine and the main drive gear reduction unit is absent from the rear of the engine. The main drive gear reduction unit has been integrated with the accessory drive unit, and is mounted in the front of the engine. The accessory drive sheaves are attached to one end of the output shaft and the other end is splined to a floating shaft passing under the engine which is coupled to the transmission mounted on the rear end of the engine. The regenerator drive is the same as that shown in Figure 27, but the regenerator seals are similar to that of Figure 28. The engine full speed RPM is 40,000.

Design Problems - The principal problem identified with the concept of Figure 28 is the exposure of the turbine nozzle actuation linkage to the hot air from the regenerator discharge. However, a gear type linkage is not required, since, the required angular movement of the nozzle vanes is only of the order of 15-20 degrees. (Reversing of the vanes for engine braking is not required, since braking is available from windmilling the compressor).

3.4.2 Preliminary Design

Because of the difficulty in meeting the 1976 Federal Standards on NO_x emission with conventional gas-turbine combustors (see reference 1), the Environmental Protection Agency requested that preliminary designs be carried out on two engines which were specifically designed for low NO_x emission. Two concepts were pursued. The first incorporates the General Electric low NO_x bypass combustor concept and the second limits combustor inlet temperature to 1000°F.

3.4.2.1 PD-1 Engine with Bypass Combustor

In this engine concept the combustor primary air is bypassed around the regenerator to reduce the primary zone combustion temperature and prevent premature ignition of the fuel-air mixture. Selection of the cycle and configuration lead to the optimization of the major design variables and ultimately to an engine design.

3.4.2.1.1 Performance Comparison of Alternate Engine Cycles

As a starting point for the selection of an optimum engine type to which to apply the bypass combustor design concept, a fuel economy comparison was made between representative variable-power-turbine free-turbine and variable-compressor and turbine single-shaft engines. Design parameter values for these engines are indicated in Table 12. The results of the comparison, shown in Figure 29, are that the single-shaft engine has better fuel mileage than the free-turbine engine, when both engines have a conventional combustor, and also when both engines have a bypass combustor. As expected, the bypass combustor engines suffer a fuel mileage penalty of approximately 30% when compared against the corresponding conventional combustor engines. Included in the performance comparison are the effects of transmission losses. For the single-shaft engine the TRACOR transmission with the efficiency values of Figure 13 were used. For the free turbine engine an automatic transmission with torque converter was used. The TRACOR transmission has lower efficiency at low speeds, but the choice of transmission does not affect the comparison.

Figure 30 and 31 provide some insight into the underlying cause of the superiority of the 3.2 pressure ratio variable-compressor and turbine single-shaft engine over the 5.0 pressure ratio variable-power-turbine free-turbine engine. Through the use of variable compressor diffuser vanes which control the engine airflow at a given speed and pressure ratio over a range of approximately 2 to 1, low power levels can be efficiently achieved at a higher speed than is possible with a fixed geometry compressor. Variation of the turbine nozzle area is required in coordination with variation of the compressor diffuser vanes in order to maintain pressure ratio and turbine inlet temperature at peak efficiency levels. Thus the operating speed range required to cover the full power range at peak achievable efficiency is reduced in the single-shaft

Table 12

Values of Engine Parameters Assumed for Comparison of Variable-Compressor and Turbine Single-Shaft and Variable-Power-Turbine Free-Turbine Engines

	Single-Shaft Engine	Free-Turbine Engine
Pressure Ratio	3.2	5.0
Inlet Temperature	85°F	85°F
Compressor Eff.	.81	.81
Cooling Flow	3%	3%
Leakage Flow	4%	4%
Gas Generator Turbine Eff.	-	.85
Power Turbine Eff.	.85	.85
Turbine Inlet Temperature	1900°F	1900°F
Regenerator Effectiveness (Full Load)	.85	.85
Combustor Eff.	.99	.99
Parasitic Power	4 HP	4 HP
Pressure Drops (Full Load)		
Inlet	1%	1%
Regenerator Cold Side	1%	1%
Regenerator Hot Side	7%	7%
Combustor	4%	4%
Turbine Diffuser	2%	2%

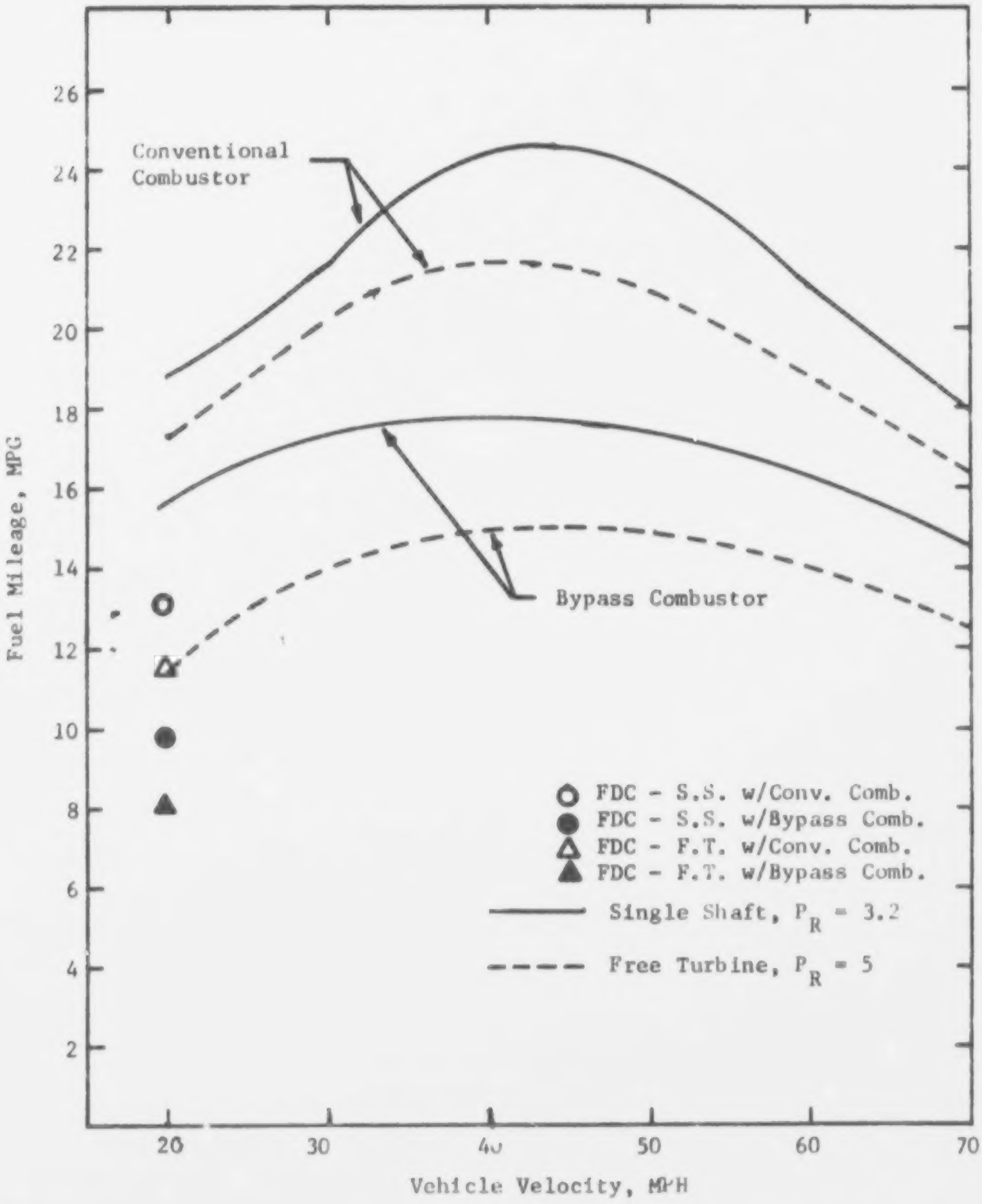


Figure 29. Comparison of Fuel Economy for Single-Shaft and Free-Turbine Engines.

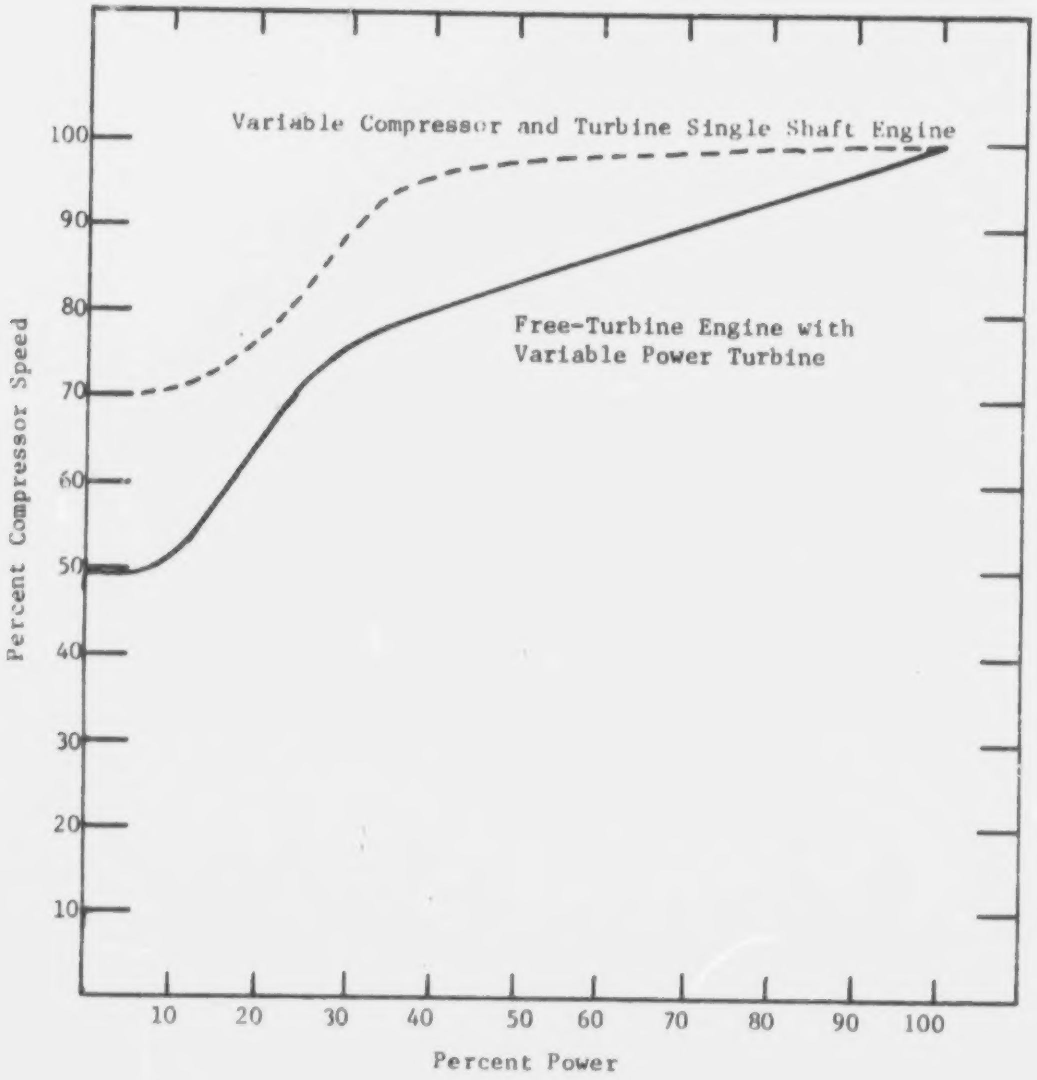


Figure 30. Comparison of Single-Shaft and Free-Turbine Rotative Speed Variations.

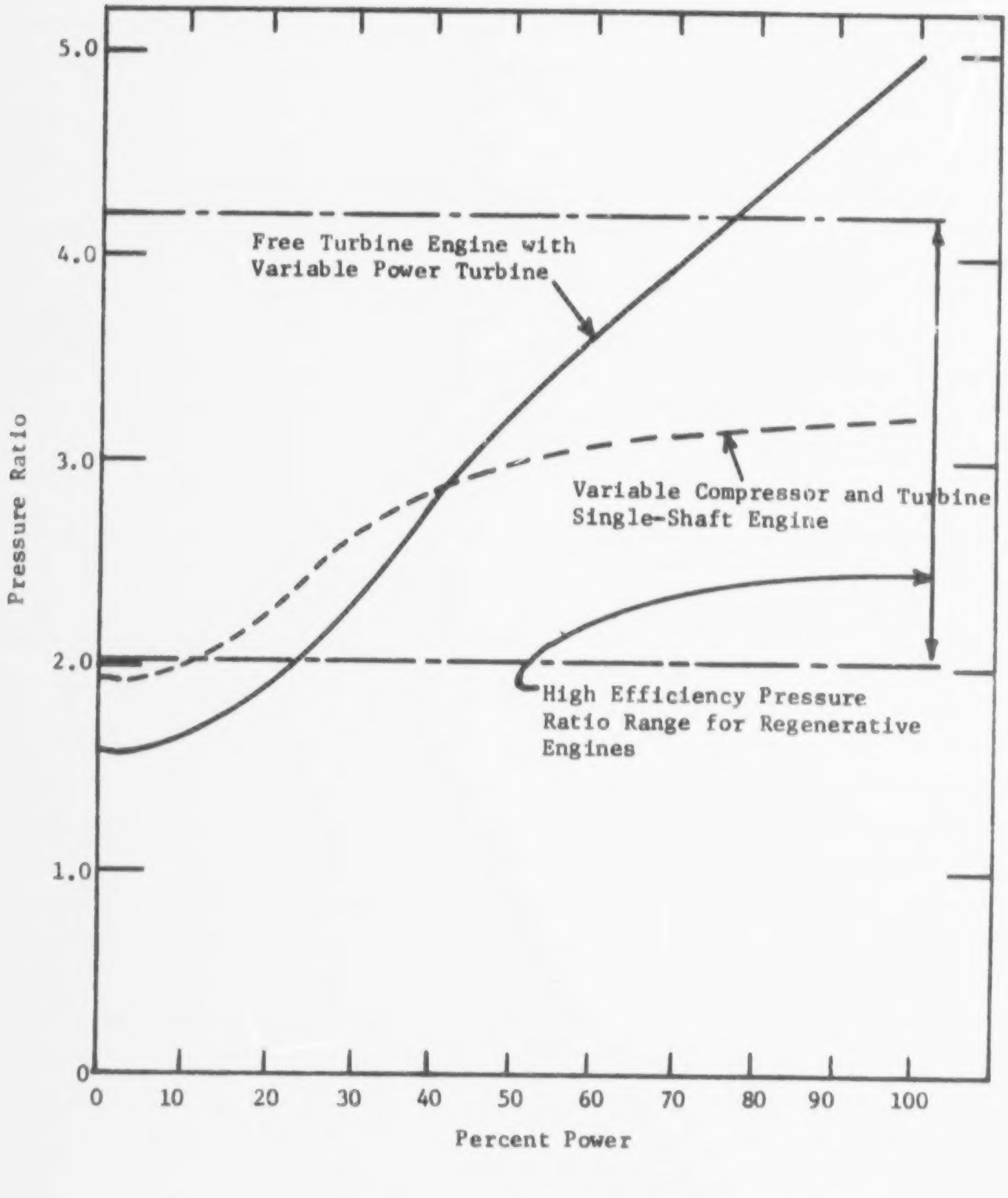


Figure 31. Comparison of Single-Shaft and Free-Turbine Pressure Ratio Variations

engine. The design goal of keeping the engine pressure ratio within the high efficiency range of the regenerative cycle over the entire operating load range is nearly achieved. From Figure 30 , it may be seen that the single-shaft variable-compressor and turbine engine operates in the 90 to 100% speed range for all power levels above 30% and that the idle speed is 70%. Although the full load pressure ratio of this engine is below that of the free turbine engine, the operating pressure ratio is higher than that of the free turbine engine at all loads below 40%. At full load the single shaft engine pressure ratio is closer to an optimum value for the regenerative cycle than is that of the free turbine engine.

A generalization of the comparison indicated by Figure 29 is shown in Figure 32 in which fuel mileage for the Uniform Simplified Engine Duty Cycle is plotted against pressure ratio for both the variable power turbine free turbine engine and the variable compressor and turbine single shaft engines. The dotted portions of these curves represent engine design conditions for which the values assumed for certain parameters require detailed investigation. In the case of the free-turbine engines, it has been found that for pressure ratios exceeding 5 very high gas generator speeds and the use of two stage compressors (transonic axial-radial) are necessary to meet the acceleration specifications at competitive engine capacity levels. As these speeds are too high for low cost ball bearings, sleeve bearings with comparatively high losses must be used. In the case of the variable-compressor and turbine single-shaft engines, compressor efficiency penalties, discussed below, must be imposed for pressure ratios above the 3.2 level as a result of design compromises involved in the achievement of high flow range. (The required flow range increases with pressure ratio and at the same time the inherent low loss flow range capability of the compressor decreases.) For these reasons the dotted portion of the curves of Figure 32 are considered to be optimistic and not representative of the presently available level of technology. Revised curves are discussed below.

Notwithstanding these reservations, the basic fuel economy superiority for the variable-compressor and turbine single-shaft engine is indicated in this figure.

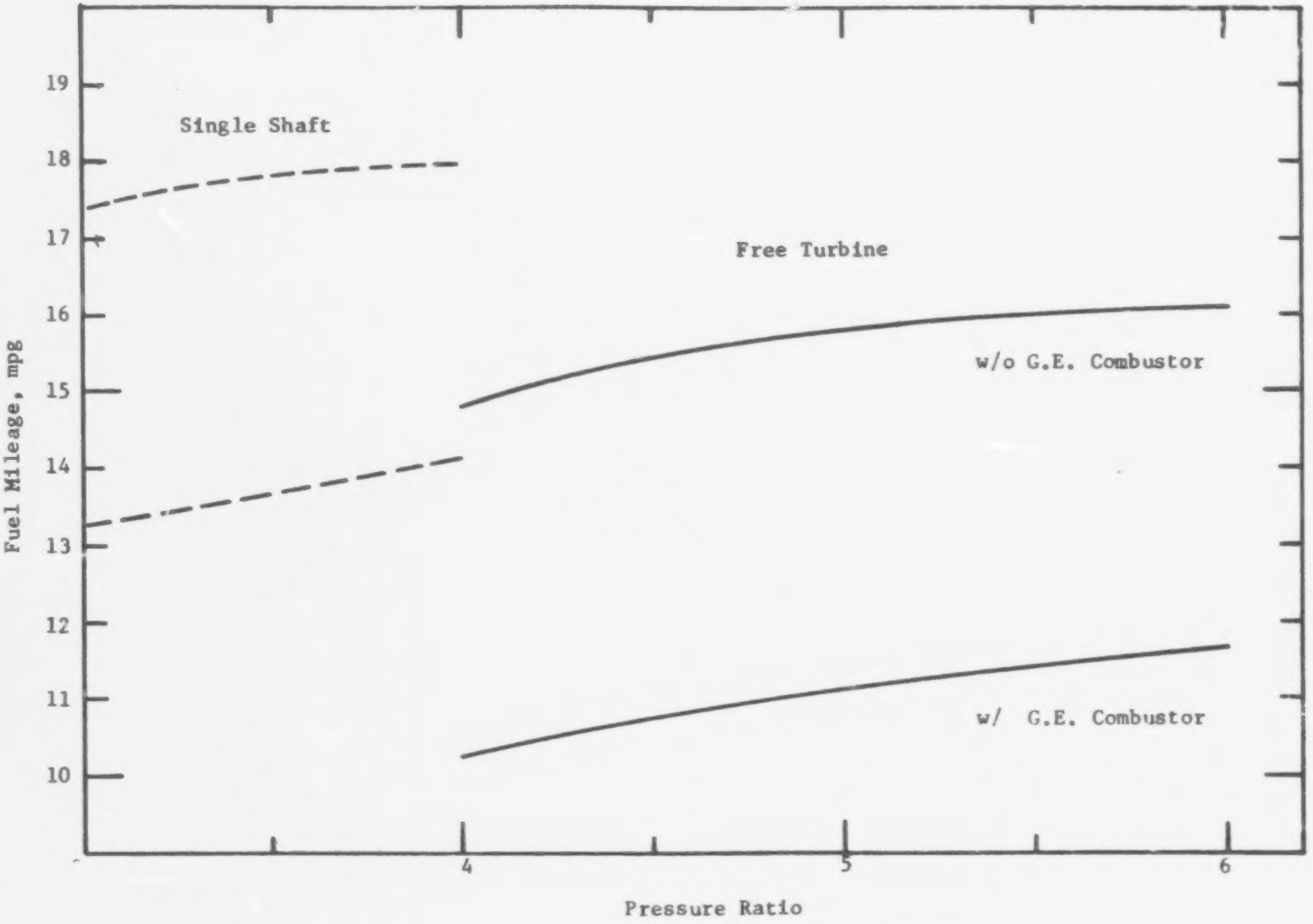


Figure 32. Comparison of Fuel Mileage of the Single-Shaft and Free-Turbine Engines for Uniform Simplified Engine Duty Cycle, Ambient Temperature, 85°F.

3.4.2.1.2 Engine Configuration Selection

As indicated above in Figure 29 the bypass combustor concept inherently involves a fuel economy penalty. To minimize this, it is important that the effectiveness of the heat exchanger be as high as possible. This strongly indicates the use of a counterflow heat exchanger of the rotating type (regenerator as opposed to a crossflow recuperator). A second consideration is the fact that the combustor concept works out most favorable with a relatively large diameter. Also, required is a division of the airflow into two streams, one of which enters the combustor from the inside and the other from the outside. Of all the possibilities considered, the engine configuration shown in Figure 33 is the most favorable for the incorporation of the bypass combustor. The combustor is located at the forward end of the engine, and the entire distance between regenerator discs is available for occupancy of the combustor structure. From the standpoint of engine compactness this is the best possible location for a combustor inherently tending to be large in diameter and short in length. An additional favorable feature of this configuration is that compressor discharge air can be readily admitted to the center of the combustor and that this air, in the process of being directed into the combustor center, can serve the useful functions of cooling the main housing and the variable turbine nozzle mechanism. It may be observed that this general configuration, which seems to be outstandingly suitable for incorporation of the bypass combustor, is natural only for the single shaft engine. A free turbine could only be accommodated if it were placed upstream of the gas generator turbine. This is not an attractive free turbine configuration. It was concluded that the single-shaft, regenerated engine, using the arrangement of Figure 33 is uniquely suited to the application of the bypass combustor, or any combustion system requiring a large envelope volume.

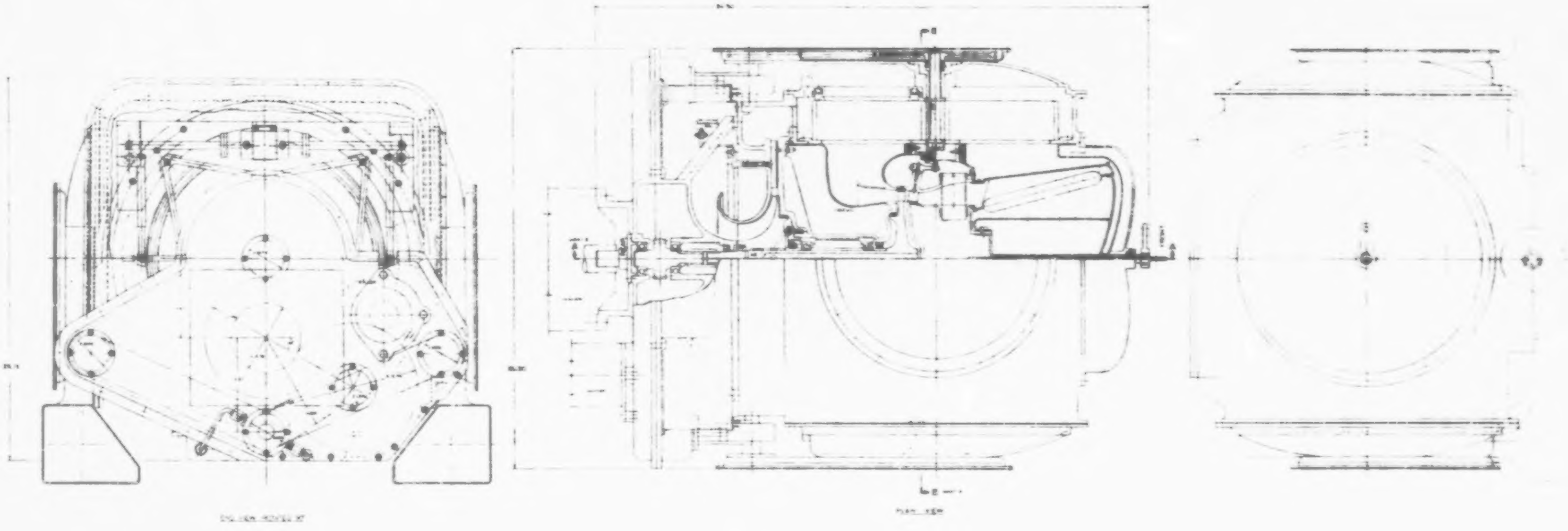


Figure 33. Single-Shaft Engine with Bypass Combustor

3.4.2.1.3 Pressure Ratio Selection

The selection of the engine pressure ratio for the PD-1 engine requires a modification of Figure 32 above, since these curves do not reflect any compressor variable-geometry-efficiency penalty related to pressure ratio. There are stong reasons to believe that, for the variable-diffuser compressor, such penalties inherently exist. A brief investigation was made to obtain a preliminary evaluation of compressor efficiency at full load and at minimum flow, idle speed, as a function of pressure ratio. This schedule of compressor efficiency vs. design pressure ratio was then introduced into cycle calculations to obtain a second iteration estimate of engine SFC at full load and at 16 HP which is the average power of the Federal Driving Cycle. Judgement was then applied to this estimate in order to select the cycle pressure ratio.

Experimental data for variable diffuser centrifugal compressors is reported in References 4 and 5 . In Reference 4 tests are reported on a radial vaned compressor designed for a pressure ratio of 2.3. Vaned diffusers with five different setting angles were employed. At constant design speed and pressure ratio, a flow variation of approximately two to one was obtained at nearly constant efficiency by successive use of all 5 vanes covering a total vane setting angle range from approximately 70° to 80°. No inlet guide vanes were used. The inducer tip incidence angle varied over approximately 25°. The inducer tip relative Mach number at maximum flow was 0.8. In Reference 5 testing of a 3.8 pressure ratio compressor using variable diffuser vanes is reported. Up to 7 to 1 flow variation at constant speed was achieved by varying the diffuser vane angle. However, marked reductions in efficiency and stall pressure ratio were observed as the vanes were closed. It was noted in Reference 5 that no attempt had been made to optimize the compressor design to accommodate variable diffuser operation.

By means of design optimization, it is possible to minimize the loss in compressor efficiency associated with diffuser vane angle flow control. It is expected that the loss in efficiency due to the use of diffuser vane controlled flow will increase with pressure ratio.

Reasons for this include the following:

1. As design pressure ratio is increased the flow Mach number at the vaned diffuser inlet increases. This effect is shown in Figure 34 . As shown in Figure 35 this results in an increase in diffuser loss ratio and also in a reduction in the flow range (as indicated by incidence angle range) between stall and choke. When variable vane settings are employed to increase the operating flow range, increased Mach number is expected to increase the loss coefficient at off optimum settings of the diffuser vanes, but the magnitude of this effect has not been evaluated. The magnitude of the diffuser inlet Mach number increase with increasing pressure ratio is amplified by the fact that impeller backslope, which reduces diffuser inlet Mach number at a given pressure ratio, is structurally feasible only at the lower pressure ratios. It is also amplified by the fact that higher inducer losses for the higher pressure ratios, discussed below, reduce efficiency and further increase the required impeller speed and also the diffuser Mach number.
2. Cycle analysis shows that the required idle to full-load airflow ratio decreases as pressure ratio increases. This effect is shown in Figure 36 for the pressure ratio range of 3.2 to 4.0. The result of a low value of this ratio is a larger range of incidence angle on the inducer, with resulting higher losses at the ends of the operating range. This effect is shown in Figure 37 , based on cascade data from Reference 19, for the pressure ratios 3.2, 3.6, and 4.0.
3. If the compressor specific speed is held constant or increased as design pressure ratio is increased the inducer relative Mach number will increase with pressure ratio. This will aggravate the effect of 2 above, since the low loss incidence angle range of the inducer decreases as Mach number increases. The minimum loss coefficient also increases with Mach number. To avoid this, a reasonable practice is to allow compressor specific speed to decrease with increasing design pressure ratio. If the design rotative speed is held constant, for example at a bearing design limit, as pressure ratio is in-

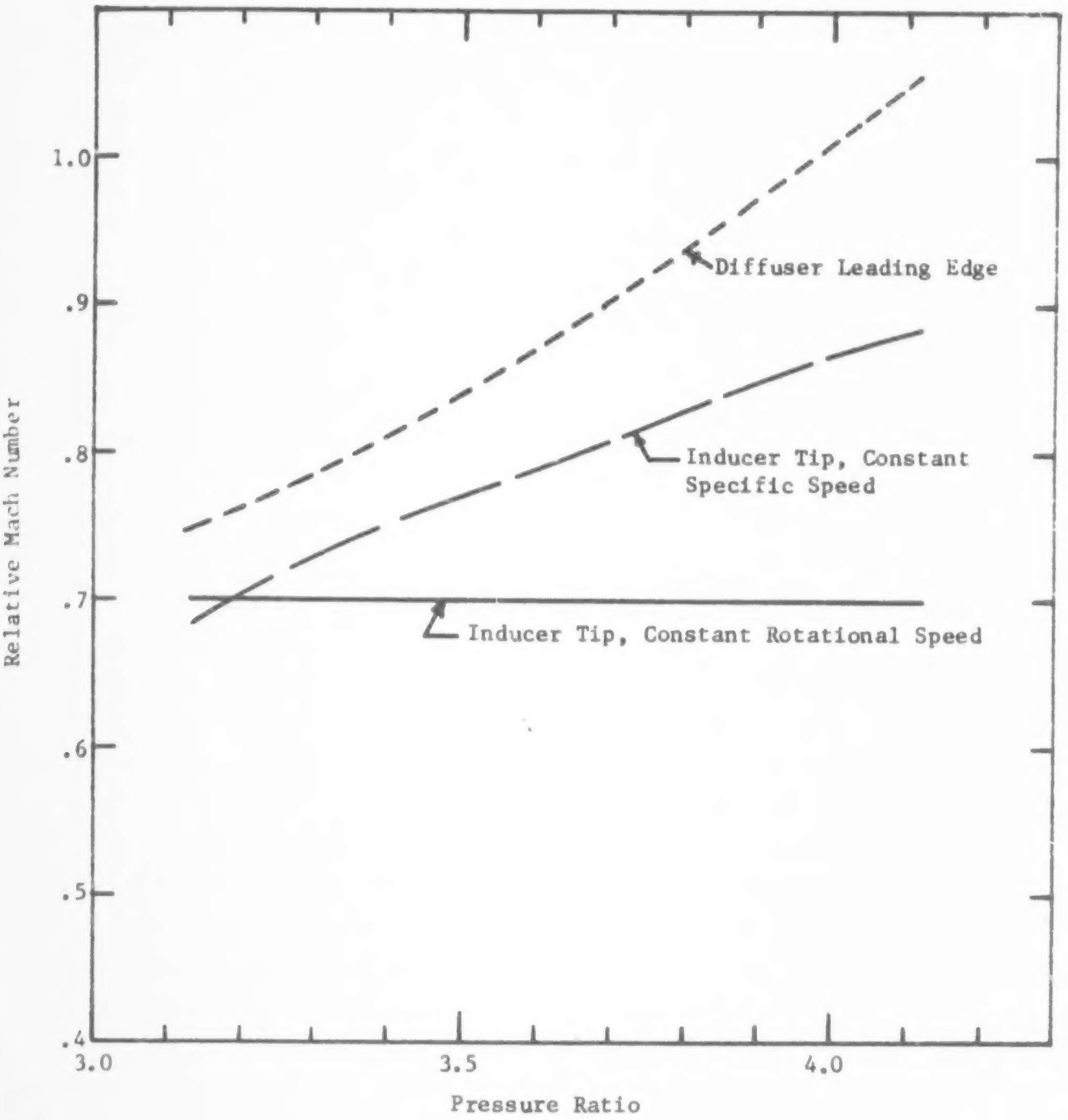


Figure 34. Variation of Mach Numbers Relative to Centrifugal Compressor Blading.

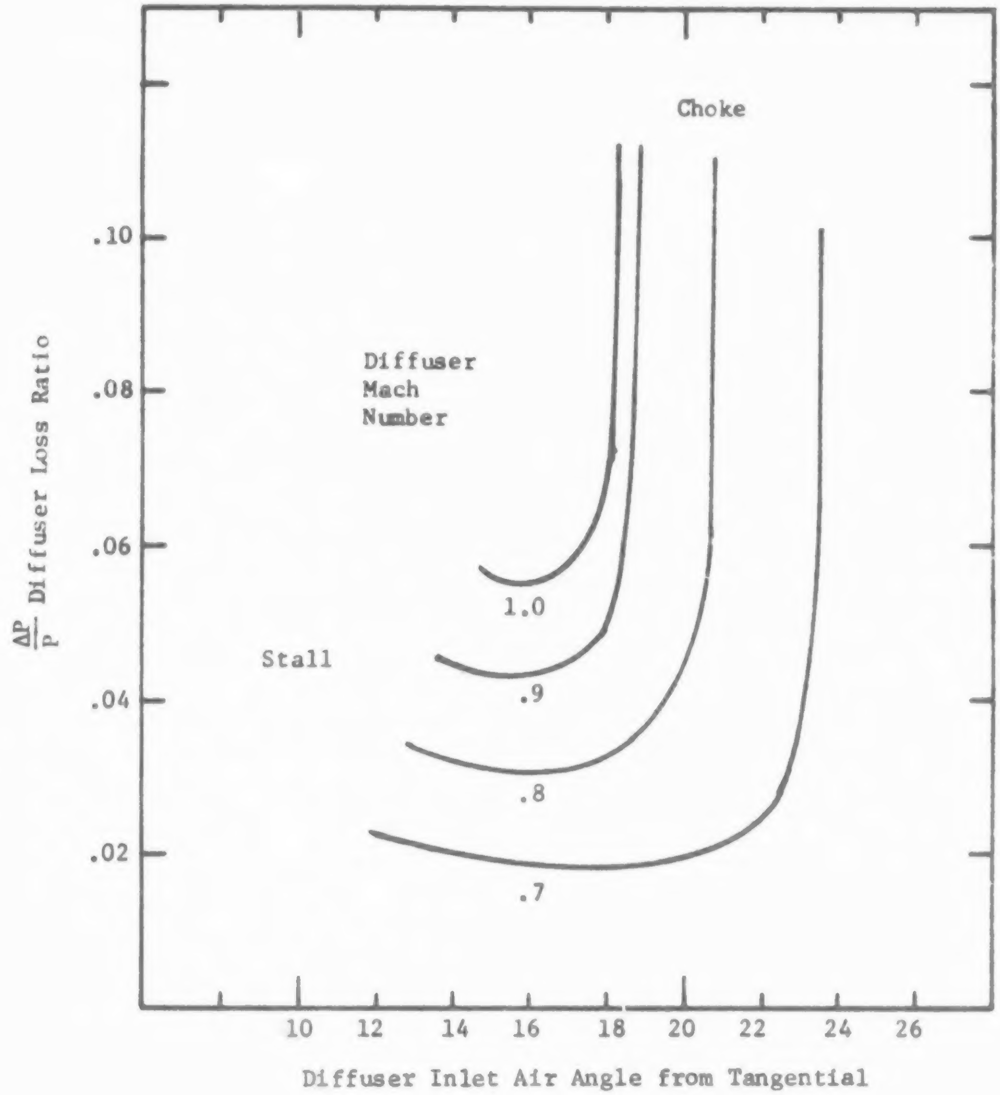


Figure 35. Variation of Diffuser Loss with Inlet Mach Number and Angle.

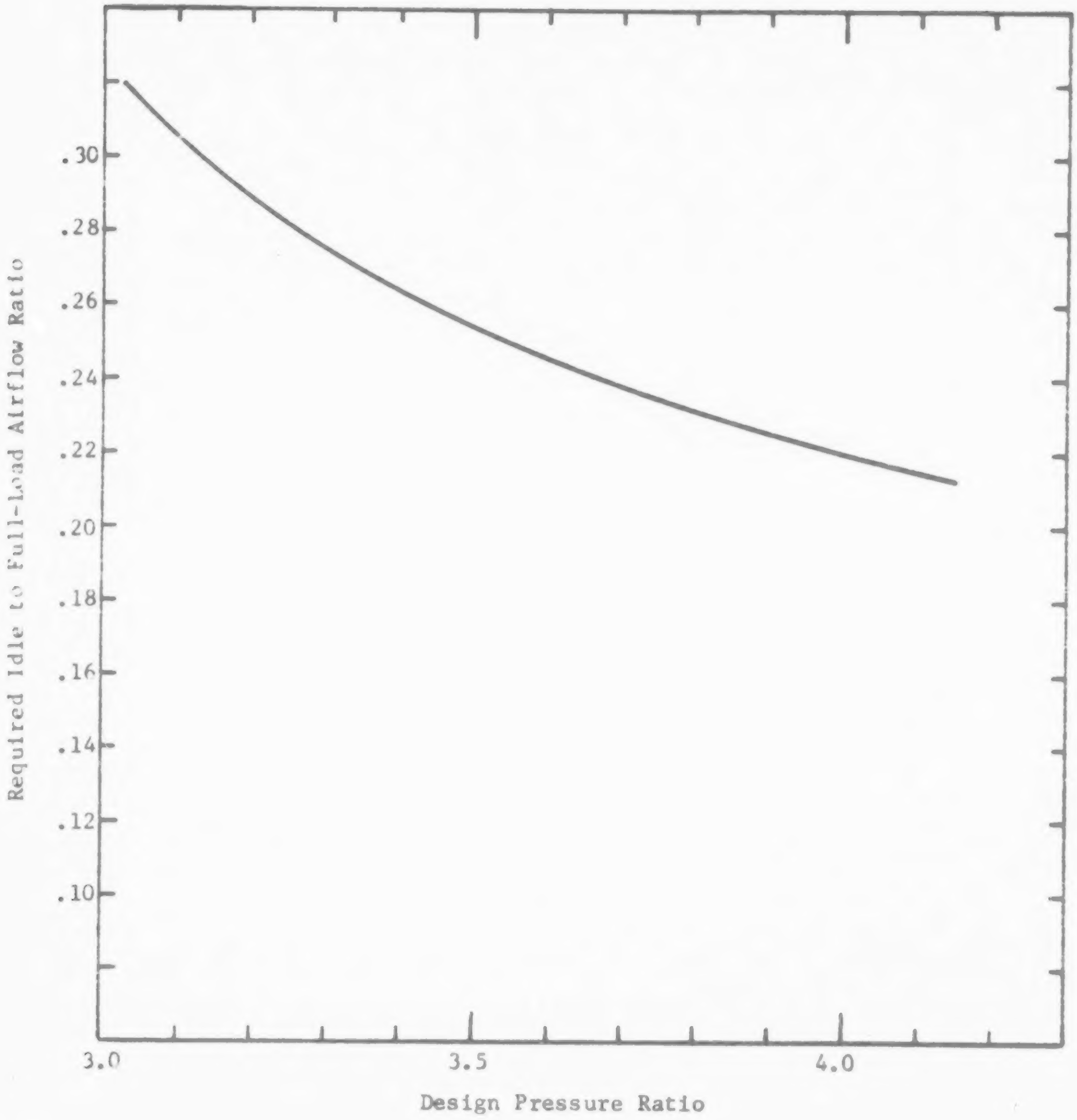


Figure 36. Variation of Idle to Full-Load Airflow Ratio with Pressure Ratio

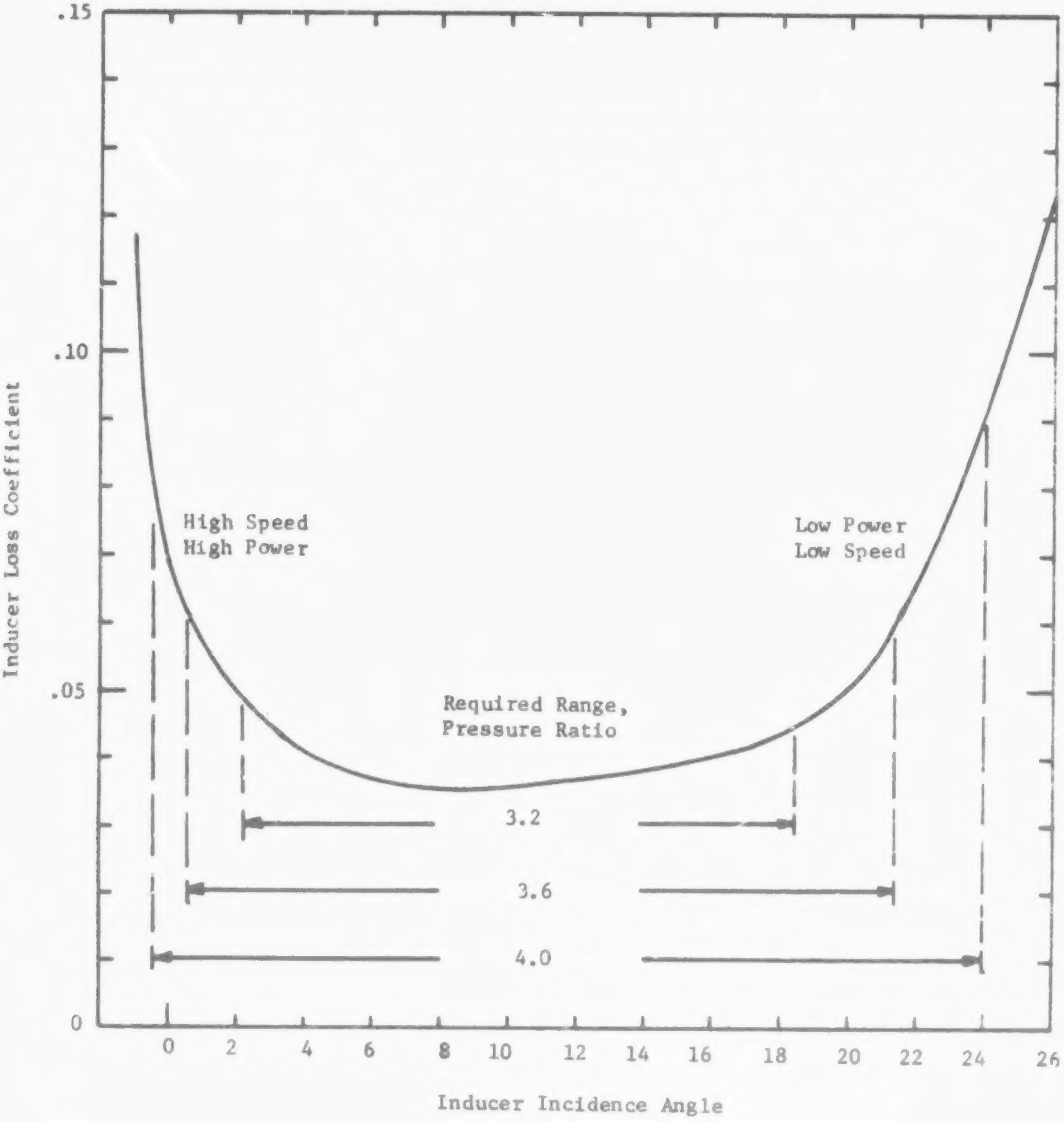


Figure 37. Variation in Inducer Loss Coefficient with Incidence Angle

creased, the inducer Mach number will remain essentially constand, and the low loss range of inducer incidence angle will also remain unchanged. This is the situation implicit in Figure 37, wherein one inducer loss coefficient-flow angle characteristic is shown for all three pressure ratios. The effect of larger compressor and turbine diameters corresponding to higher pressure ratios and lower efficiency will be to increase the rotor moment of inertia and will result in a larger maximum power capacity to meet acceleration specifications. This, in turn, will increase the required flow range of the compressor and aggravate the effect of 2 above.

A preliminary evaluation of the effects mentioned above has resulted in the estimated compressor efficiency variation with design pressure ratio for both fixed and variable geometry compressors shown in Figure 38. These curves indicate a variable geometry penalty which increases with pressure ratio, particularly at full speed where the inducer and diffuser Mach numbers are at maximum values and the inducer incidence angles are below optimum. A smaller penalty is shown for the low speed low flow condition, where inducer incidence angles are above optimum, but Mach numbers are low. At intermediate conditions, such as high speed with vanes practically closed, the penalties should be less than shown.

The effects of the compressor pressure ratio efficiency penalties on the engine specific fuel consumption at high and low power levels and also on the full power airflow are shown in Figure 39 as functions of design pressure ratio in the range of 3 to 4.0. It will be noted that the curves are quite flat, with specific fuel consumption and full load airflow showing minimums at approximately 3.6 pressure ratio. Although the curves of Figure 39 seem to indicate an optimum pressure ratio of 3.6, the indicated potential gains for this pressure ratio over the selected value of 3.2 are small (3-1/2% in SFC at 16 HP and essentially zero at 150 HP, 4% in full power airflow). At the 3.2 pressure ratio level the designers freedom to introduce features such as impeller backslope, which are conducive to low loss penalties, without encountering stress limits and/or high rotor inertia are substantially greater than at the 3.6 level. Therefore, the value of 3.2 was selected as the design

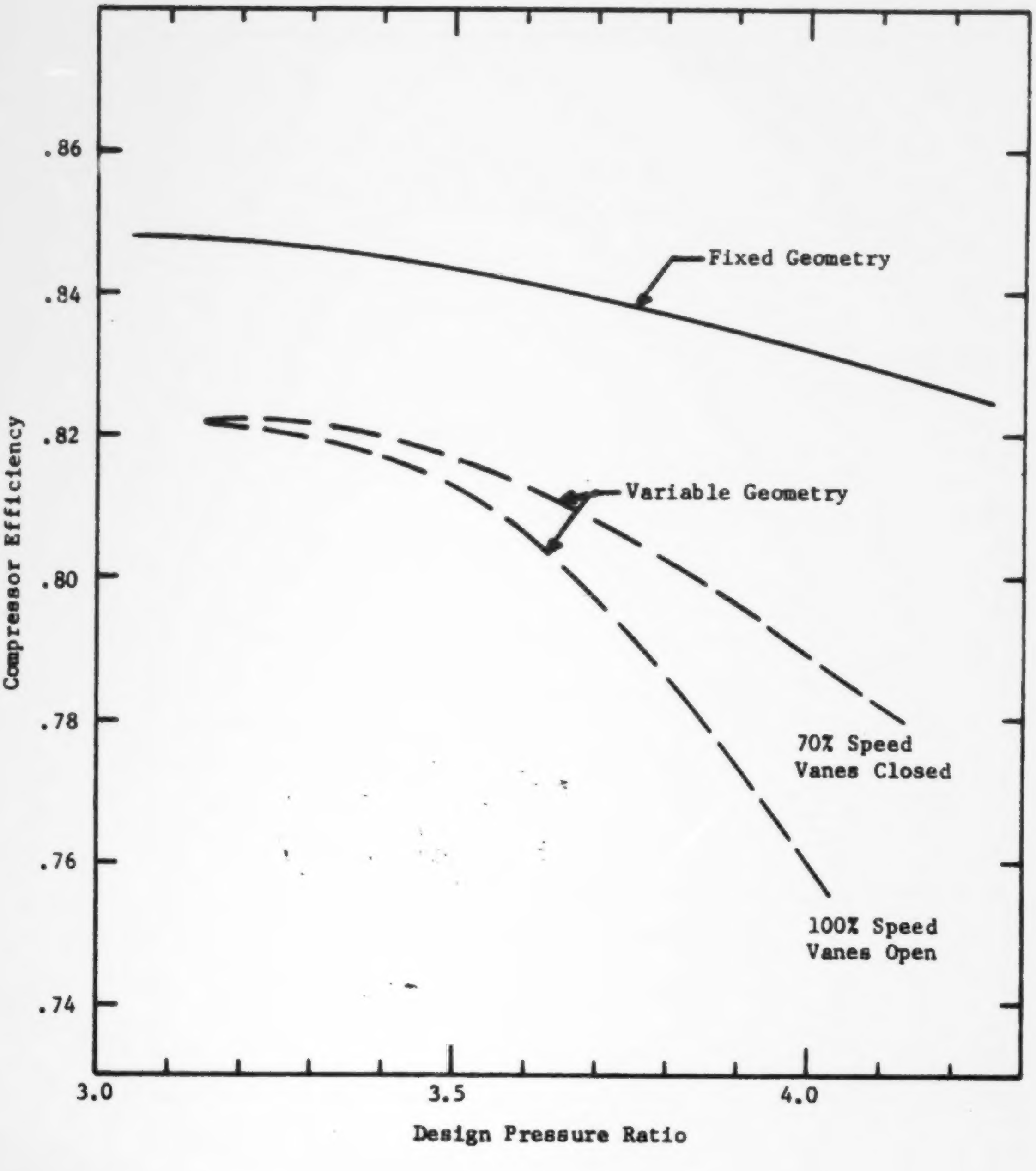


Figure 38. Variation of Compressor Efficiency with Pressure Ratio

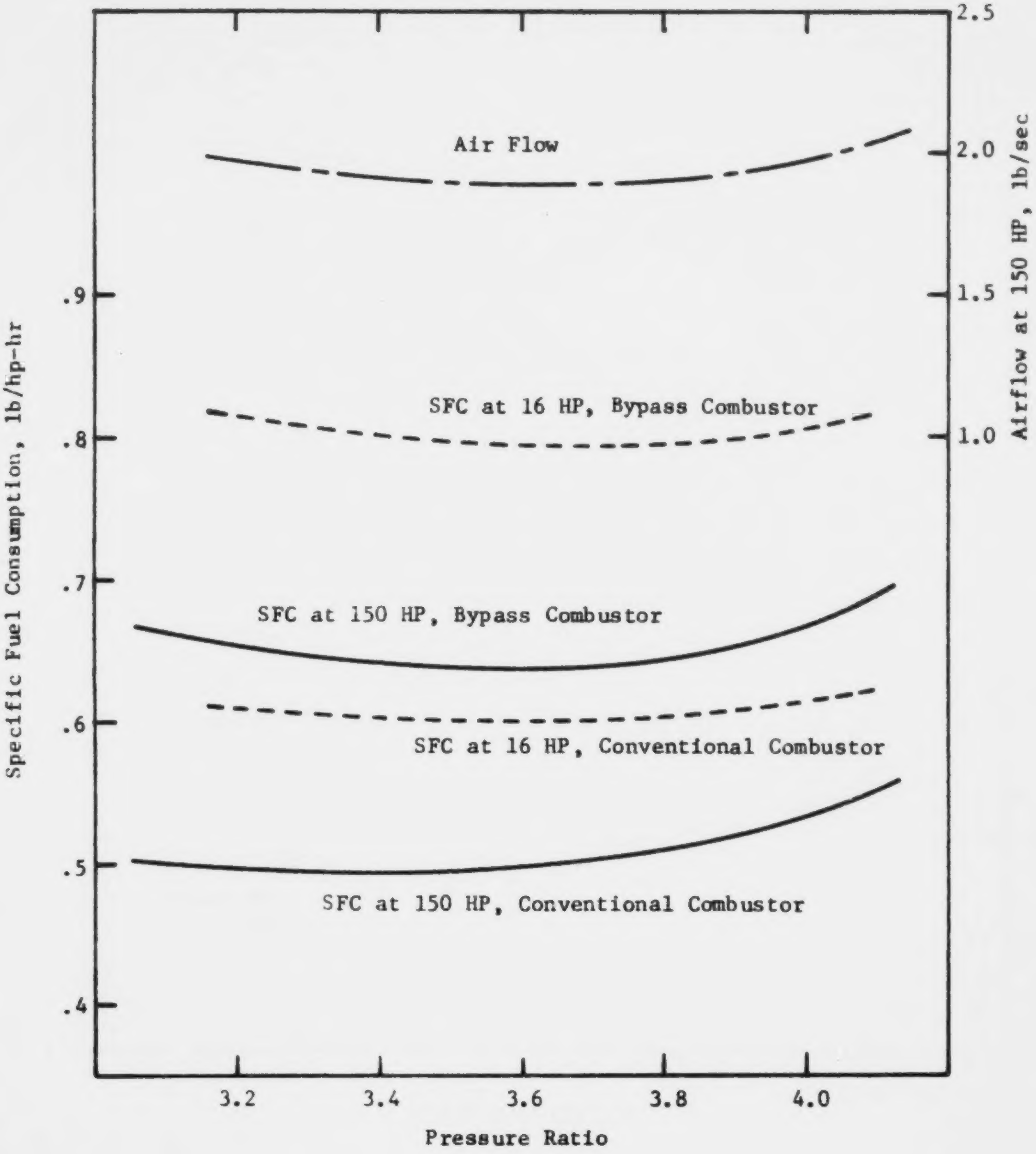


Figure 39. Effect of Design Pressure Ratio on Airflow and Specific Fuel Consumption

pressure ratio for PD-1 because it is believed that at the present state of the art, the potential gains for higher pressure ratios do not justify the increased design risk. An experimental program is required to assess the potential gains of a higher design pressure ratio.

3.4.2.1.4 Rotative Speed Selection

Selection of the engine full power rotative speed was based upon four major considerations: bearing design, engine acceleration characteristics, turbine rotor stress and compressor design.

Bearings - A major design goal is the minimization of parasitic losses, since at low power levels these losses have a large effect on fuel economy. In keeping with this goal, it was desired to use anti-friction bearings instead of sleeve bearings because of the large percentage difference in losses (factor of 3 to 5 depending upon detailed design features) for the two bearing types. At the same time it was recognized that the rotative speed of ball bearings must be limited in order to achieve the necessary life in a commercial grade (class 5) bearing. High precision aircraft type bearings were ruled out on the basis of high cost. Bearing manufacturers were consulted, and it was found that a 30 mm ball bearing could be employed under a 300 lb. thrust load with a B_{10} (statistically no more than 10% failures) life of 1000 hours at a speed of 33,000 RPM. Also a 25 mm bearing could be employed with a 200 lb. thrust load with the same B_{10} life at a speed of 40,000 RPM. The latter bearing type was selected for the final PD-1 engine design. The speed of 40,000 was judged to be the highest speed at which class 5 ball bearings could be employed with high reliability under the imposed thrust load.

Acceleration Characteristics - One of the advantages of the variable-geometry single-shaft engine over the free-turbine engine is that the acceleration specifications can be met with a given power capacity at a substantially higher value of design-point rotor kinetic energy than that permitted for the free-turbine-engine gas generator. This results from the fact that a large accelerating torque is made immediately available at the start of an acceleration by opening the vanes and increasing fuel flow without any increase in rotative speed. Despite this inherently favorable feature of the single-shaft engine, acceleration performance can be improved by reducing design-point kinetic energy. To a first approximation the moment of inertia varies as the 4th power of the compressor and turbine rotor diameter and for a fixed pressure ratio these diameters vary inversely with speed. Therefore, design-point kinetic energy varies inversely with the square of the speed. High speed is therefore favorable for acceleration. It was found that the rated power of the engine can be reduced approximately five HP by increasing the full load engine speed from 32,700 to 40,000 RPM and meet the same acceleration specifications. Comparative acceleration performance values are shown in Table 13. These Figures indicate that a 55% reduction in rotor moment of inertia (34% reduction in design-point kinetic energy) provides only a small improvement in acceleration performance. There appears to be little to gain from this standpoint by increasing speed above 40,000 RPM. This is a feature unique to the single-shaft variable-geometry engine.

Turbine Rotor Stress - Centrifugal stress at the turbine blade root is proportional to the square of the rotative speed and to the first power of the blade annulus area. The first iteration engine design was optimized at a rotative speed of 32,700 RPM, since this permitted the use of CRM-6D, a 5% nickel alloy, for the turbine rotor. A high cooling airflow (5%) was required however, and the turbine inlet temperature was limited to 1850°F. The turbine rotor was heavy (13 pounds) but contained a very small amount of high cost material (nickel). In order to improve fuel economy and acceleration performance the rotative speed of the second iteration design was raised to 40,000 RPM. Turbine cooling air was reduced from 5% to 2%, and the turbine inlet temperature was increased to 1900°F. Turbine rotor weight decreased to 7.5 pounds but the material was changed to IN 713 having approximately 70% nickel. The engine total rotor moment of inertia was reduced from .027 lb ft sec^2 to .012 lb ft sec^2.

Compressor Design - In section 3.4.2.1.3 above the importance of compressor inducer relative Mach number to variable diffuser performance has been mentioned. In order to maintain a low level of this parameter a relatively low specific speed (77 in RPM, cfm and ft units) or less is required since this permits a low ratio of inducer diameter to impeller diameter, and a low level of inducer blade speed while holding down the

Table 13

Comparison of Acceleration Performance Between 32,700 RPM (0.028 lb ft sec^2 inertia) and 40,000 RPM (0.012 lb ft sec^2 inertia) Single-Shaft Engines

	32,700 RPM Design	40,000 RPM Design
Distance Travelled in -		
1 sec, ft	4.3	5.1
5 sec, ft	140	154
10 sec, ft	501	523
Velocity After 13.5 sec, mph	66	67

air inlet Mach number. A low inducer relative Mach number permits a high tolerance to incidence angle variation, and a minimization of inducer losses and of overall impeller losses resulting from a positively or negatively stalled inducer. From the standpoint of compressor efficiency, limitation of rotative speed to a value of 40,000 RPM is favorable.

3.4.2.1.5 Idle Speed Selection

The PD-1 engine idle speed has been set at 70% full engine speed. This selection is based on the following considerations:

Acceleration Performance - Acceleration performance of the PD-1 engine is quite sensitive to the idle speed selection. This results not only from the fact that at a higher idle speed the required rotor speed and kinetic energy change required between idle and full power is reduced, but also from the fact that the power immediately available for acceleration without change of rotor speed upon opening vanes and increasing fuel flow is substantially increased. As shown in Figure 40 the available power approximately doubles between 60% and 70% speed. It is estimated that the rated power can be reduced approximately 10 HP without loss in acceleration performance by increasing the idle speed from 60% to 70%.

Fuel Economy at Low Power Level - As idle speed is increased the low power level pressure ratio is increased. The low power airflow must be decreased in order to maintain a given level of SFC as idle speed is increased. This means an increased airflow range and required incidence angle range on the compressor inducer. However, the fuel economy penalty resulting from the bypass combustor decreases with increasing pressure ratio, and with decreasing turbine inlet temperature. At a power level of 12 HP the engine specific fuel consumption is 14% lower for a 70% idle speed than for a 60% idle speed. At the 5 HP level, however, this trend is reversed, the 70% idle speed specific fuel consumption being 6% higher than the 60% idle speed. On balance, it has been estimated that no significant fuel economy penalty results from increasing the idle speed from 60% to 70%.

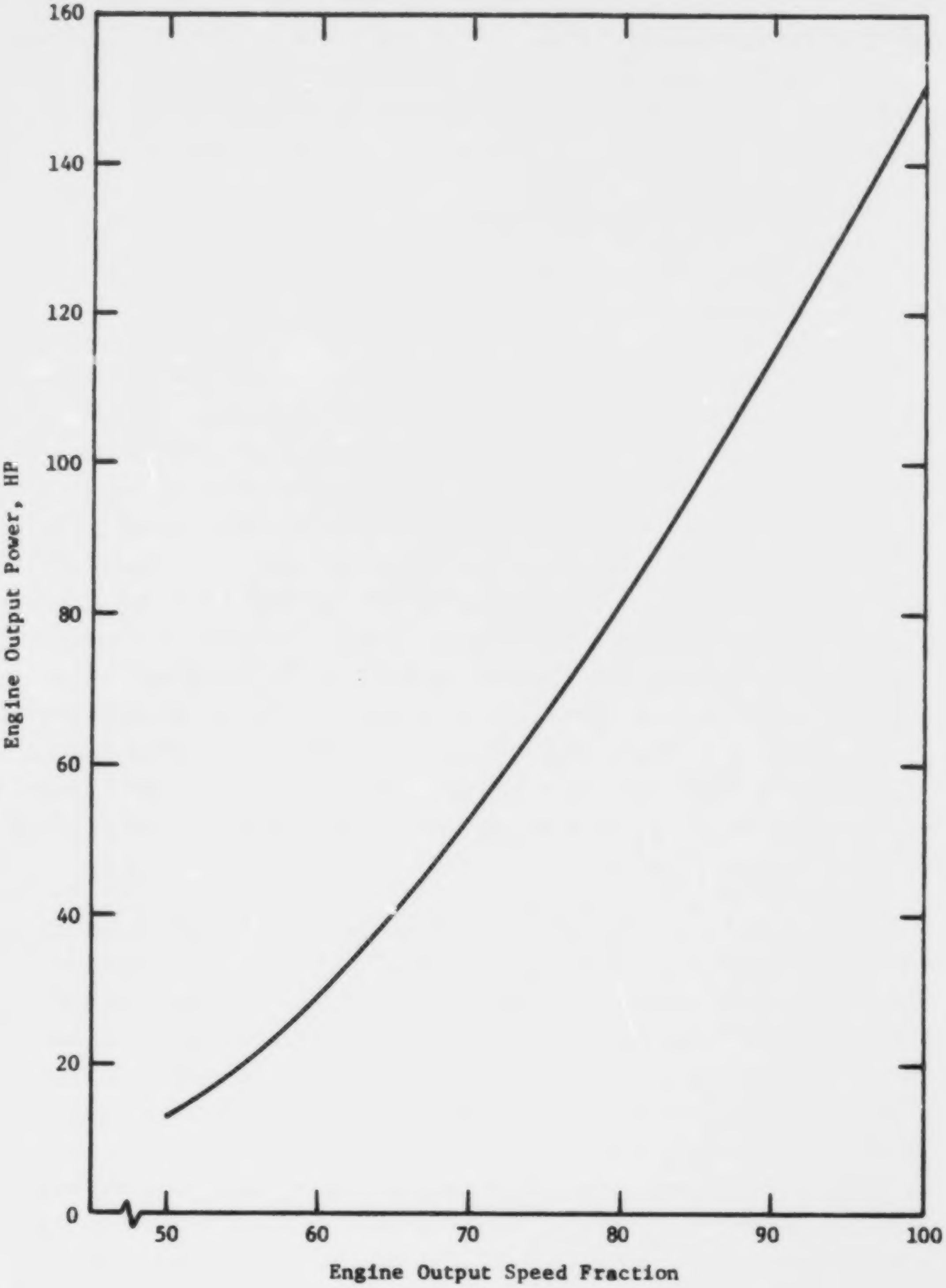


Figure 40. Variation of Maximum Power with Engine Rotative Speed for the Single-Shaft (PD-1) Engine, 105°F Ambient Temperature

Noise - A high idle speed is subject to criticism on the basis of noise. In the case of the PD-1 engine, however, the 100% rotative speed impeller tip speed is low because of the low design pressure ratio. Thus the impeller tip speed of the PD-1 at 70% idle condition is about the same as that of a 6:1 pressure ratio engine at 52% idle speed. The absence of inlet guide vanes is also favorable to a low noise level.

3.4.2.1.6 Description of Engine Design

General - Layout drawings of the variable-geometry single-shaft PD-1 engine are shown in Figures 41, 42, and 43.

The engine is designed for front mounting in a vehicle. The gear case and transmission are at the rear, as is the air inlet. The combustor is located at the front of the vehicle. The arrangement provides for engine accessories to be mounted on the combustor outer casing at the front of the engine. These are driven from a shaft passing under the engine, which is driven through gears and a chain from the (single) engine shaft. The regenerator discs are mounted and housed at the sides of the engine. These discs are centrally driven through a worm gear and chain drive from the accessory drive shaft. Overall dimensions of the engine, shown on Figure 41 are 25-1/2 inches overall height from bottom of exhaust ducts, 34-1/2 inches long, and 26-1/2 inches wide. Engine weight is 484 pounds. The figure is based on 150 HP output at 105°F inlet temperature, and is subject to some reduction corresponding to a final power sizing of 134 HP at 105°F.

In Figures 41 , 42 and 43 the engine flow path can be traced through the inlet filter enclosing the upper half of the space between the main engine housing and the gear case, into the compressor, through the variable diffuser vanes to the compressor discharge scroll. At the sides of the engine are openings in the main housing through which compressor discharge air enters the space enclosing the regenerator discs. There are two outlet paths for airflow from this space. One path, taken by the primary combustion air, is ducted into the space between the outer and inner containment of the main housing and from there into the plenum space surrounding the turbine nozzle actuating linkage through passages located at the top and bottom of the engine (90° from the regenerators).

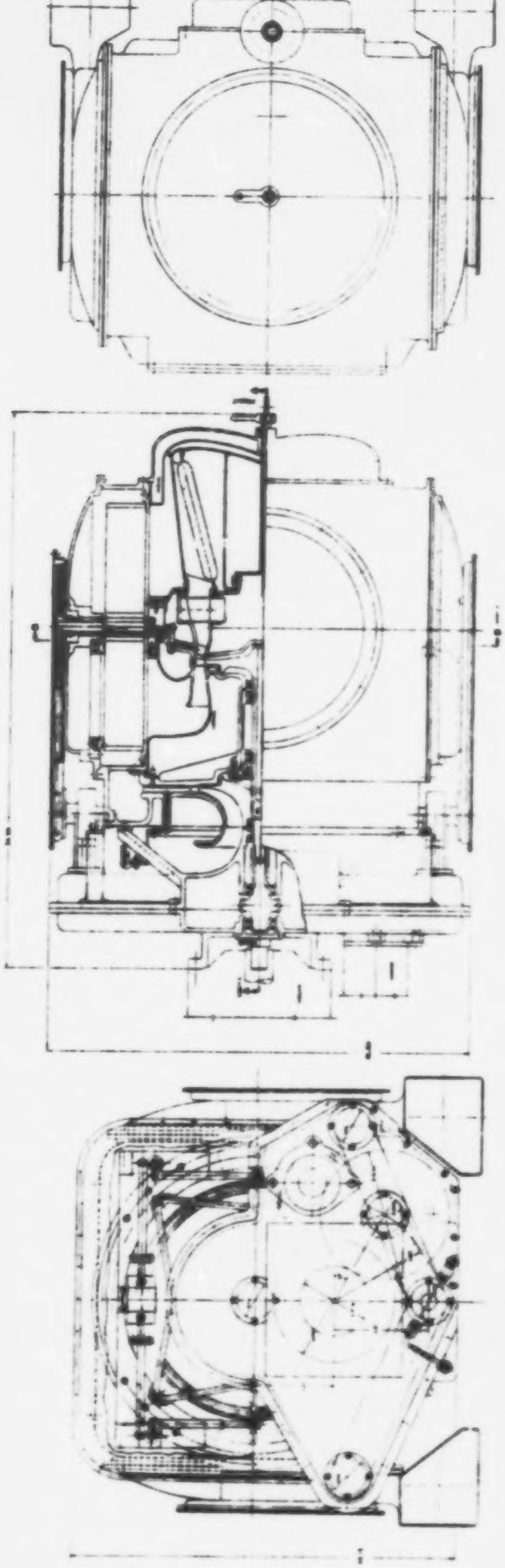
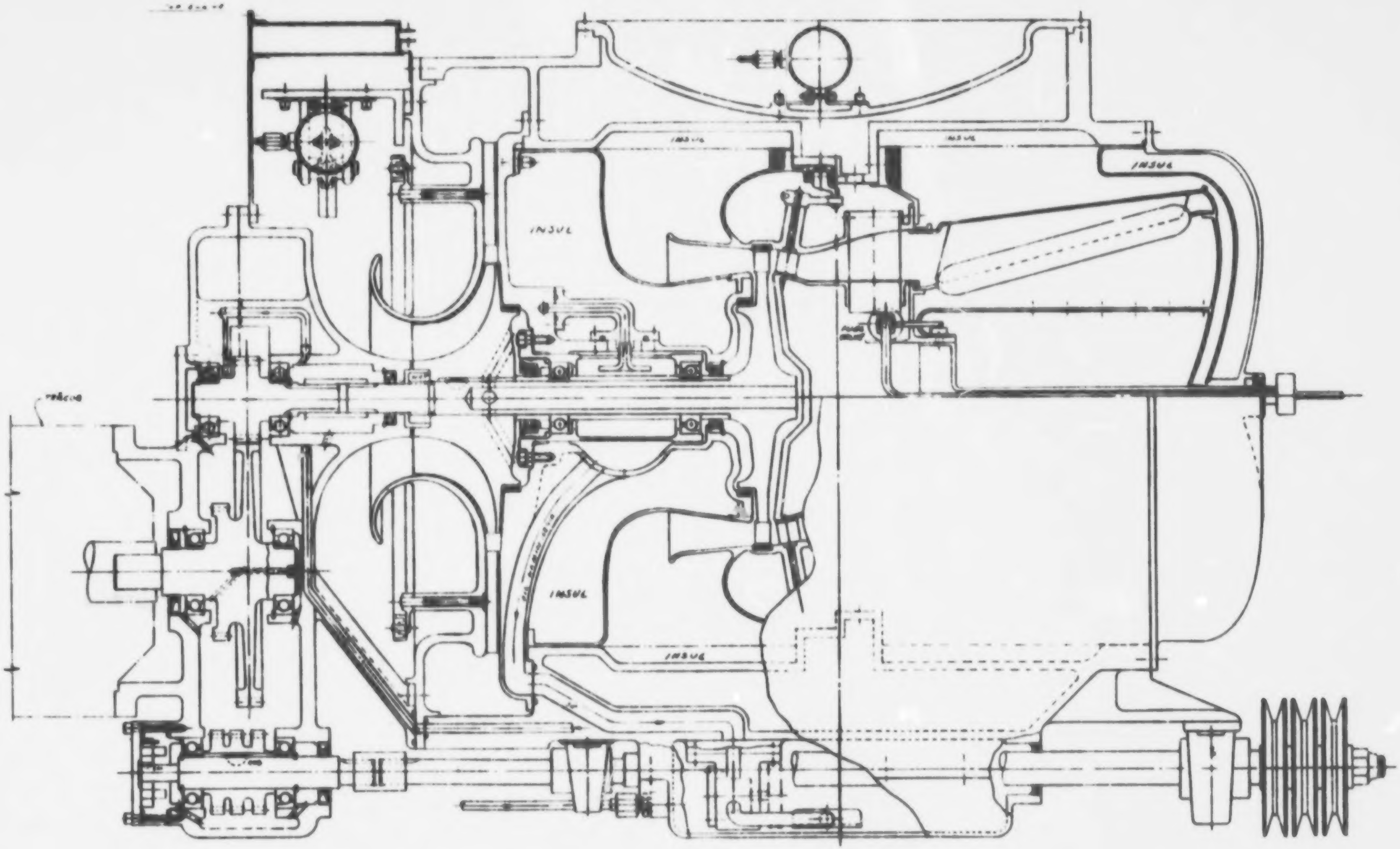


Figure 41. Single-Shaft Engine With Bypass Combustor.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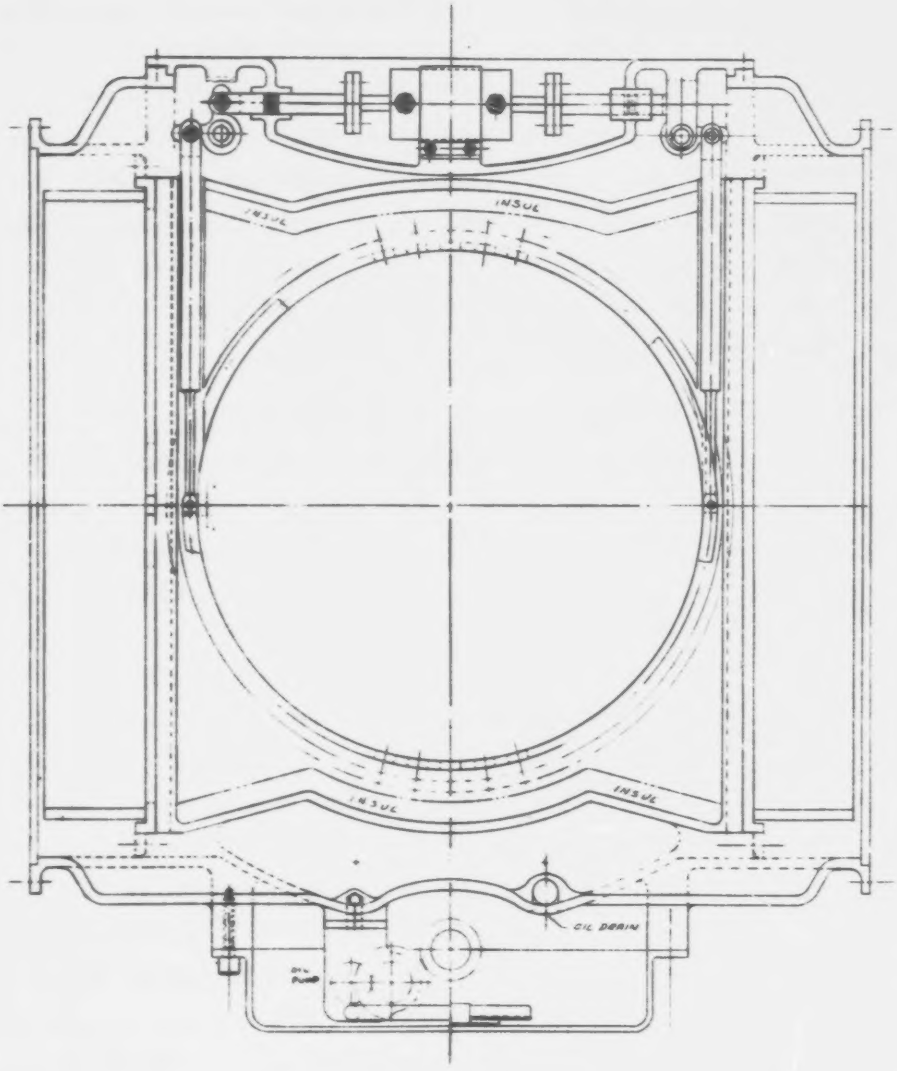
SECTION- AA

Figure 42. PD-1 Engine Drawing. Section AA.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SECTION - BB

Figure 43. PD-1 Engine Drawing. Section BB.

From the actuating linkage space this air passes through the hollow struts supporting the inner body shell which forms the inner wall of the turbine annulus flow path and into the inside of the bypass combustor structure. The excess air of the cycle passes through the halves of the regenerator discs furthest from the compressor and into the combustor from the outside. After mixing in the combustor the two air flows pass through the turbine together, and, from the turbine discharge diffuser and discharge plenum, pass through the regenerator halves closest to the compressor and out the exhaust ducts.

Structurally the engine is integrated around the main housing. This is presently designed as a nodular iron casting weighing 140 pounds. Possibilities exist for the redesign of this part as an aluminum casting, which would effect a substantial weight saving. The relatively low pressure and temperature of the compressor discharge air, to which the uninsulated housing surfaces are exposed, are factors favorable for the success of such a redesign. Mounted on the main housing are the rotor support frame, which carries the two shaft bearings, the compressor shroud plate upon which are mounted the variable diffuser vanes, the regenerator covers which carry the regenerator rotor bearings, drive sprockets and upper seal rings, the top access cover, upon which is mounted the turbine nozzle actuator cylinder, the oil sump housing, the accessory drive shaft bearing supports, and the combustor outer cover. Also attached directly to the main housing are the turbine shroud diffuser assembly onwhich mounts the variable turbine nozzle vanes, and the turbine discharge inner flow boundary shell. The gear casing, which is a split aluminum casting, is mounted on the compressor shroud plate. Also attached to this shroud plate is the support bracket for the diffuser vane actuating cylinder and associated linkage.

The drive gear pinion is mounted on two bearings and driven through a spline coupling from the engine shaft. One stage of gear reduction is required to drive the TRACOR transmission at a maximum input speed of 10,000 RPM. The accessory drive shaft is driven from the transmission drive gear through one stage of gear reduction and a final chain drive. Maximum accessory shaft speed is 4,000 RPM. Chain drive from the accessory gear shaft is used to drive the regenerator worm shafts. The regenerator worm gear assemblies are attached to the outside of the gear casing. The starter is mounted on the outside of the gear casing. The starter pinion engages the accessory drive gear. An overrunning clutch is employed to prevent reverse torque from driving the starter after engine startup.

The engine rotor is mounted on two 25 mm ball bearings. These are class 5 (commercial grade) bearings. The turbine bearing, which is an angular contact type, carries the rotor thrust load. In the selected arrangement the turbine thrust load adds to the compressor thrust load. It is therefore necessary to employ turbine and compressor seals to control the net thrust load to the value of 200 pounds which is a limiting thrust load on comfortable bearing life margin at 40,000 RPM. (The estimated B_{10} life under these conditions is 1000 hours). Space inside the compressor impeller seal is vented through the hollow shaft to the forward side of the turbine disc. This seal leakage flow purges hot gas from the space between the turbine disc and the opposing baffle plate. The space inside the turbine rotor seal is pressurized with compressor discharge air. Leakage through this seal purges hot gas from the outer downstream side of the turbine disc. The turbine bearing is flexibly mounted on a "squirrel cage" structure. Vibration of this structure is damped by the introduction of oil in the small clearance between the bearing mounting sleeve and the bearing support frame.

Oil flow for the lubrication of all bearings is supplied from a gear pump located in the bottom sump driven from the accessory drive shaft. A drain passage is provided from the bottom of the bearing support frame to the main sump. A scavenge pump is also provided at the bottom of the gear case for return of oil to the main sump.

Materials - Critical materials selection problems involve the following components:

Turbine rotor (casting)
Turbine nozzle vanes
Hot static parts
Compressor rotor (casting)
Frames and housing (castings)

The first design iteration turbine rotor material was CRM-6D. This design, shown in Figure 44 , has a maximum shaft speed of 32,700 RPM.

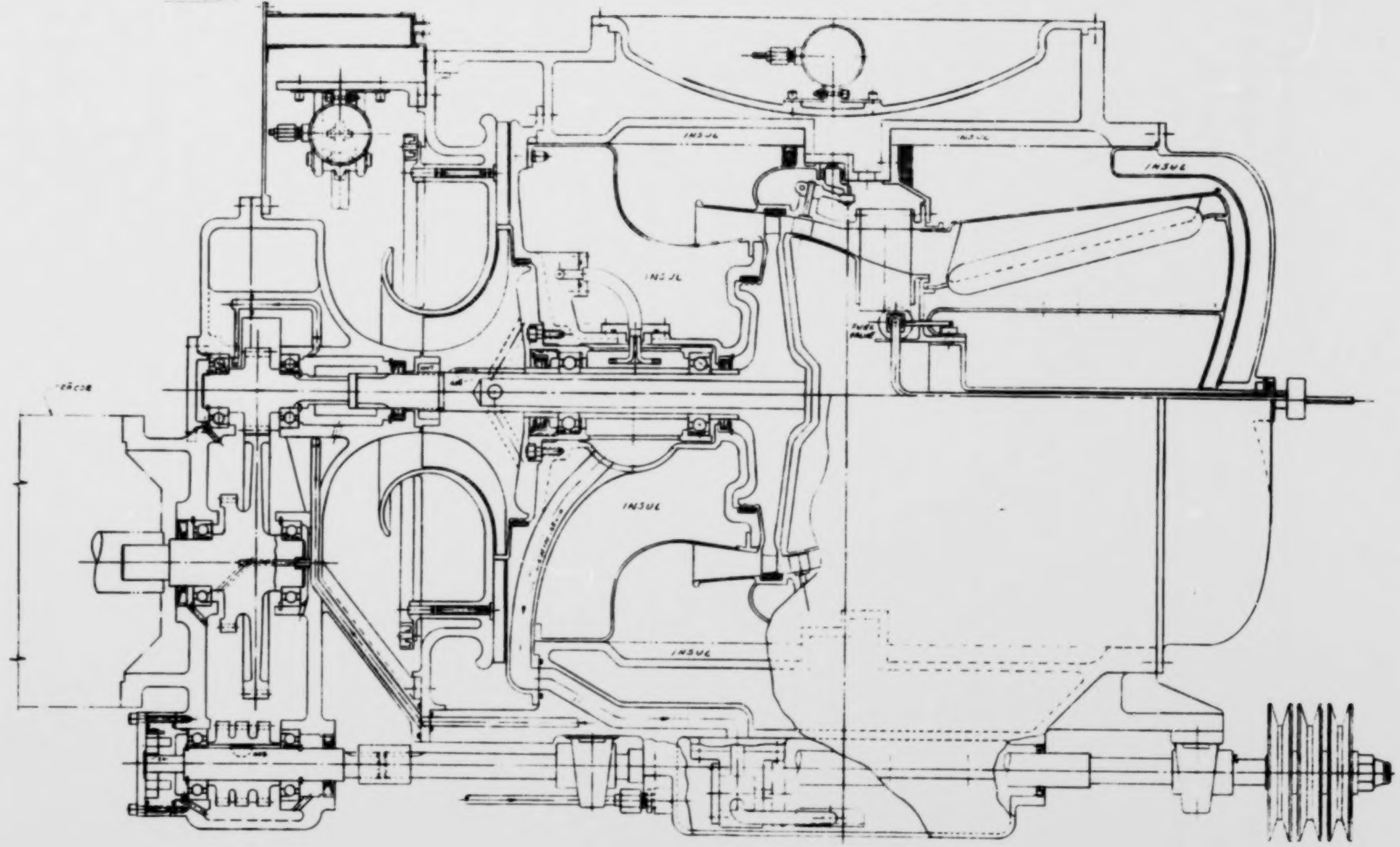
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Figure 44. PD-1 Engine Drawing With CRM-6D Turbine Rotor. Section AA.

Turbine inlet temperature is limited to 1850°F. Five percent cooling air is required. Rotor inertia is relatively high. The turbine rotor nickel weight is quite low (.6 lb).

To improve fuel economy and acceleration performance the second iteration design of Figure 42 was defined. This has a shaft speed of 40,000 RPM and a turbine inlet temperature of 1900°F. Two percent cooling air is required. Rotor moment of inertia is reduced by a factor of .43. The material is IN 713LC. The turbine rotor nickel weight is 5 lb.

The selected turbine nozzle vane material is silicon nitride. This has the advantage of low cost and no strategic material content. Thermal stresses are minimal in the variable nozzle design with individually cantilevered vanes. Foreign object impact is very improbable in this engine with inlet filter and regenerator. The attachment of the lever arms for actuation of the vanes is a design problem that will need attention.

Hot static parts operating above 1500°F are Hastelloy X. Those operating below 1500°F are 304 stainless steel.

The compressor rotor is 355 aluminum alloy. At the 3.2 pressure ratio this alloy has sufficient strength to permit some impeller vane backslope (at least 30°). This substantially improves the efficiency.

Aluminum is employed for the compressor shroud plate and the gear casing. The present engine design employs nodular iron for the main housing and the rotor bearing support frame. This specification may be over-conservative. However, more thorough temperature and stress analysis is necessary on those parts in order to determine the feasibility of substituting aluminum. Aluminum would effect substantial weight reduction and possibly some cost reduction.

Engine Installation - Figures 45, 46, and 47 show three views of the engine installed in a standard six-passenger automobile. These figures indicate that this automobile provides more than adequate space for containment of the engine.

3.4.2.2 PD-2 Engine with 1000°F Combustor Inlet

In this engine concept the combustor inlet temperature (out of the heat exchanger) is limited to 1000°F to facilitate the development of a

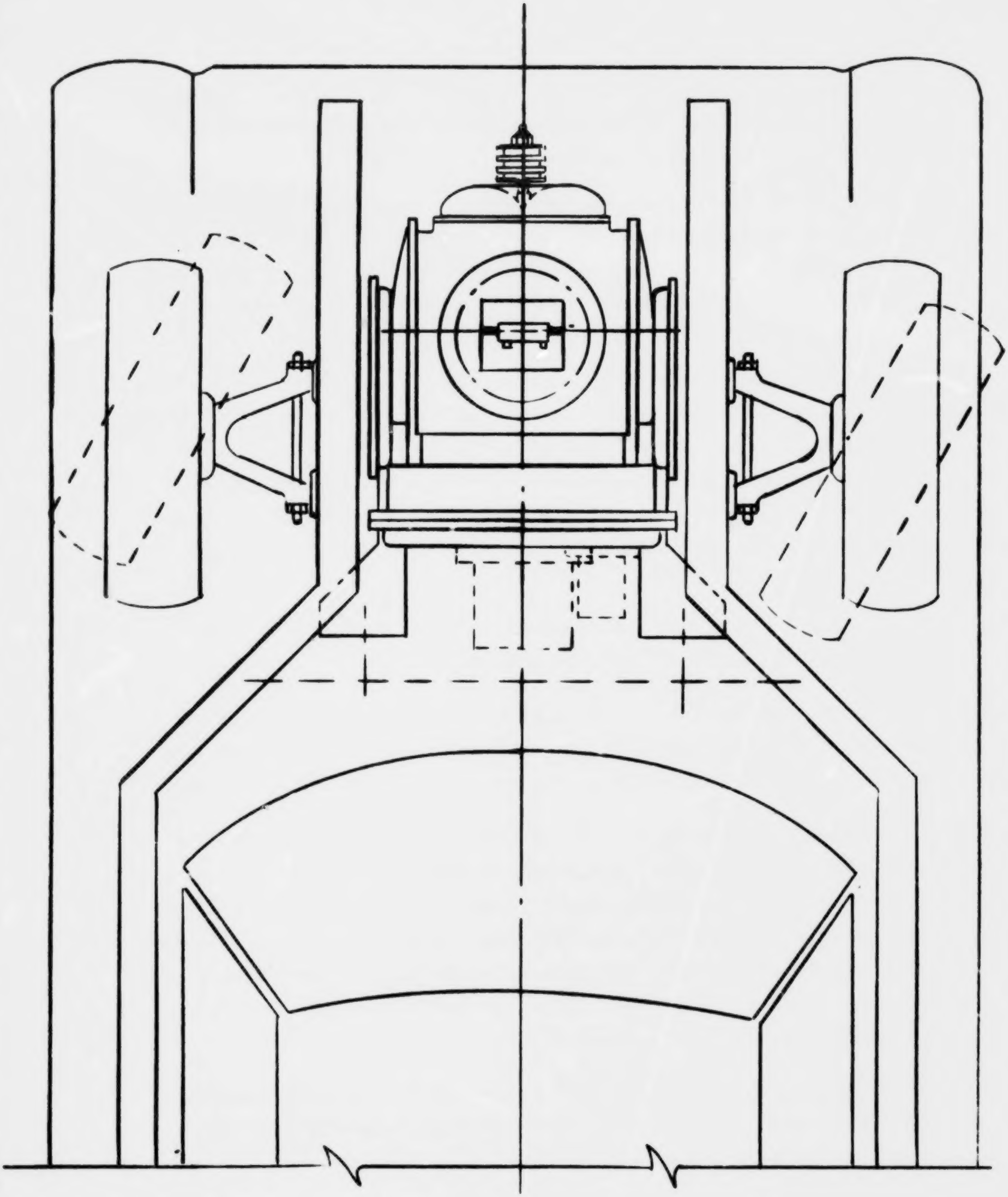


Figure 45 PD-1 Engine Installed in a Standard Six-Passenger Sedan. Top View.

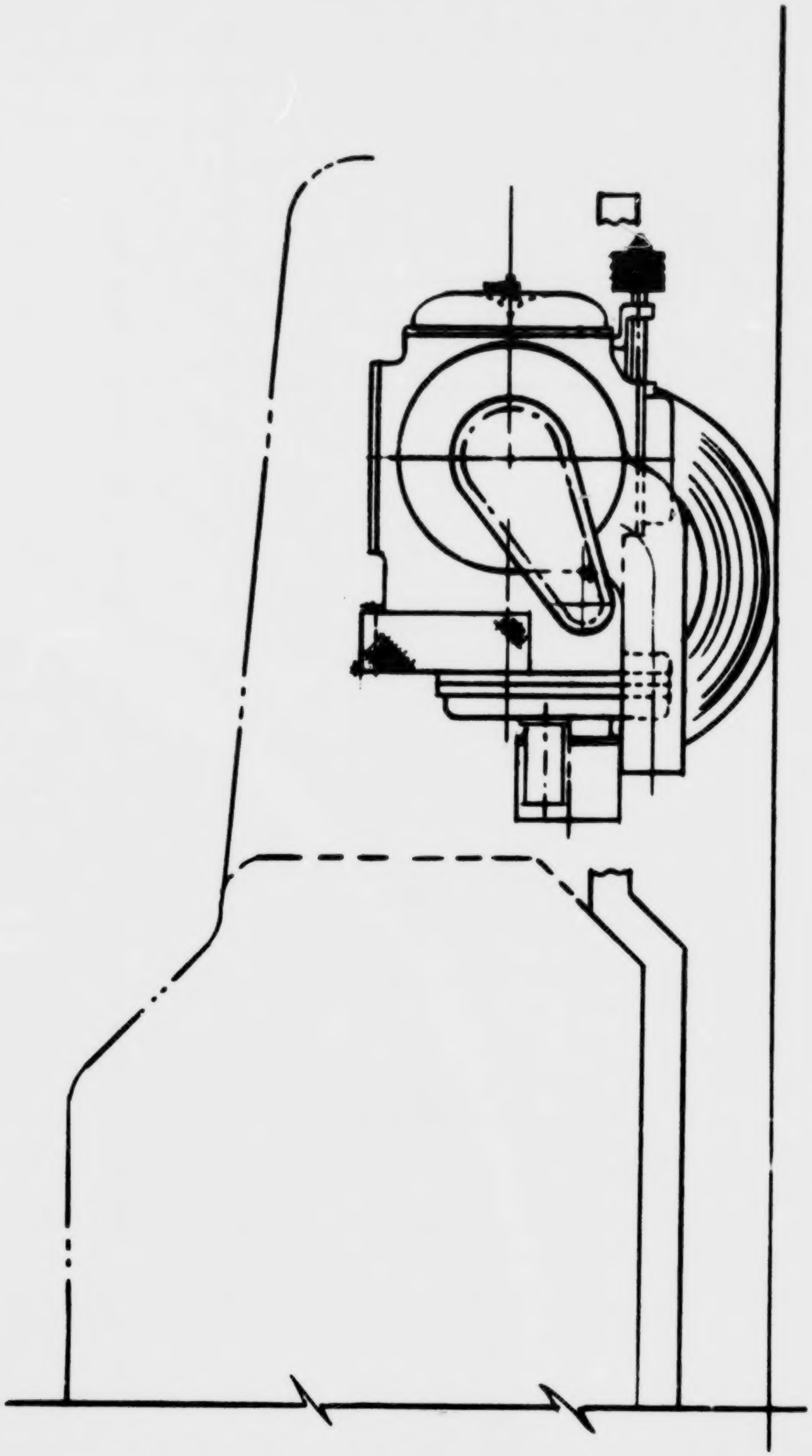


Figure 46. PD-1 Engine Installed in a Standard Six-Passenger Sedan, Side View.

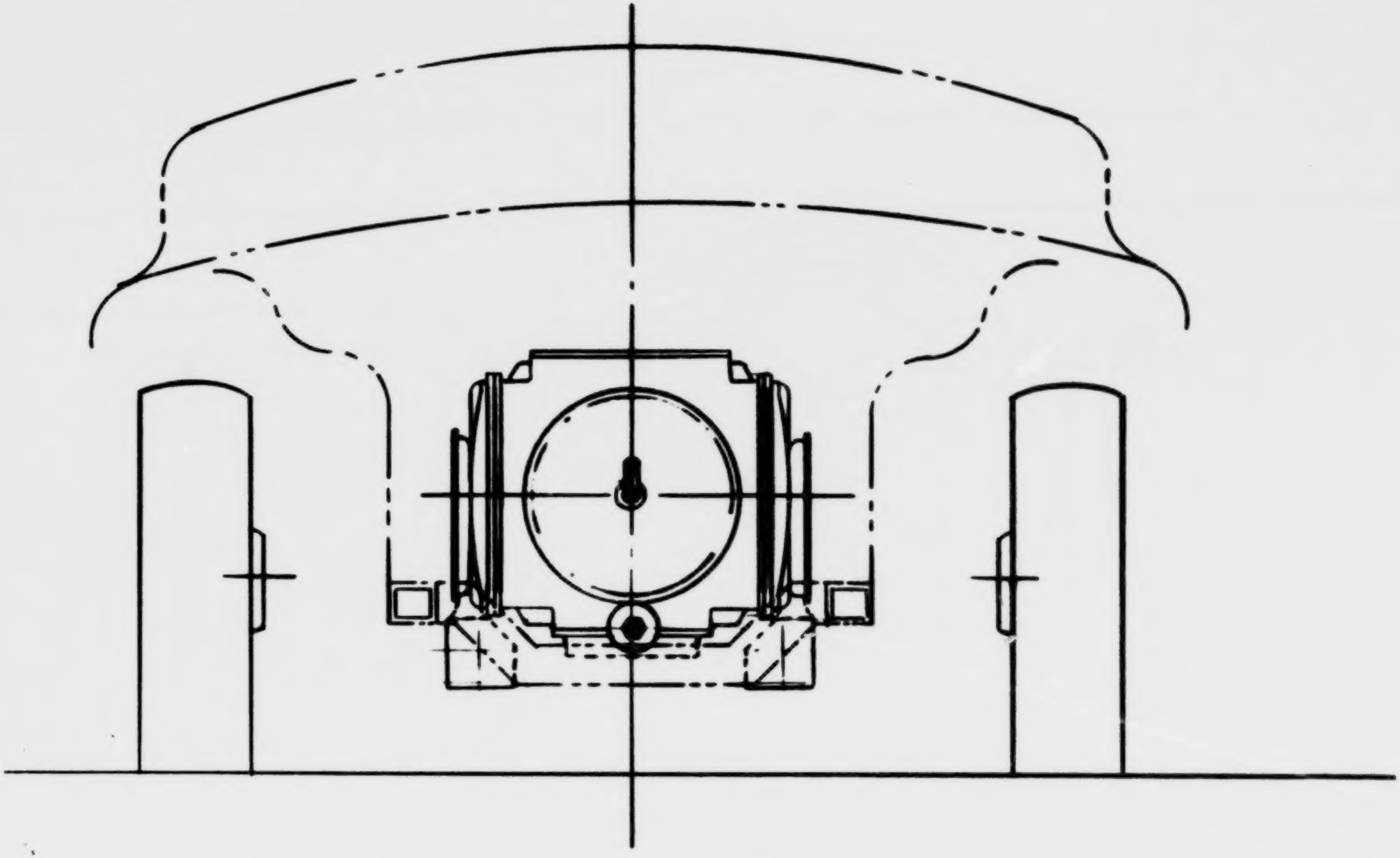


Figure 47 PD-1 Engine Installed in a Standard Six-Passenger Sedan. Front View.

low-NO_x combustor. Optimization of the cycle, selection of the type of heat exchanger, rotative speed and pressure ratio lead to an engine design.

3.4.2.2.1 Cycle Parameter Optimization

In Figure 48 the variation of engine specific fuel consumption as a function of design pressure ratio is shown for full load power (150 HP) and for 8.5% power. For each pressure ratio the heat exchanger effectiveness is set at a value such that for a turbine inlet temperature of 1900°F and an inlet air temperature of 105°F the combustor inlet temperature is 1000°F. The heat exchanger design point effectiveness is plotted as a function of design point pressure ratio in Figure 48. At the design point specific fuel consumption increases very slowly with pressure ratio in the range of 6 to 8 and then increases more rapidly above a pressure ratio of 8. At the low power level there is a steady decrease of specific fuel consumption with pressure ratio between 6 and 8. Performance at the low power level was calculated by a free turbine (variable power turbine) engine off design computer program. Above a design pressure ratio of 8.3 the low power specific fuel consumption increases because the turbine inlet temperature must be reduced in order to avoid low speed compressor stall. On this basis a pressure ratio of 8 is an optimum value. The corresponding value of heat exchanger effectiveness is .78. These values were selected for the PD-2 engine.

3.4.2.2.2 Selection of Cycle Type and Engine Configuration

Because of the relatively high pressure ratio required by this engine the concept of a single shaft engine with a variable diffuser compressor was discarded as being unworkable in accordance with reasoning presented above. Of all the cycle concepts evaluated in the early phases of the program the most suitable one for this engine is the free-turbine engine with variable power turbine. This cycle was selected.

Because of the relatively low heat exchanger effectiveness required, it was recognized that a counterflow heat exchanger of the rotating type is not required and that a stationary cross flow recuperator provides adequate effectiveness and further would offer the following advantages:

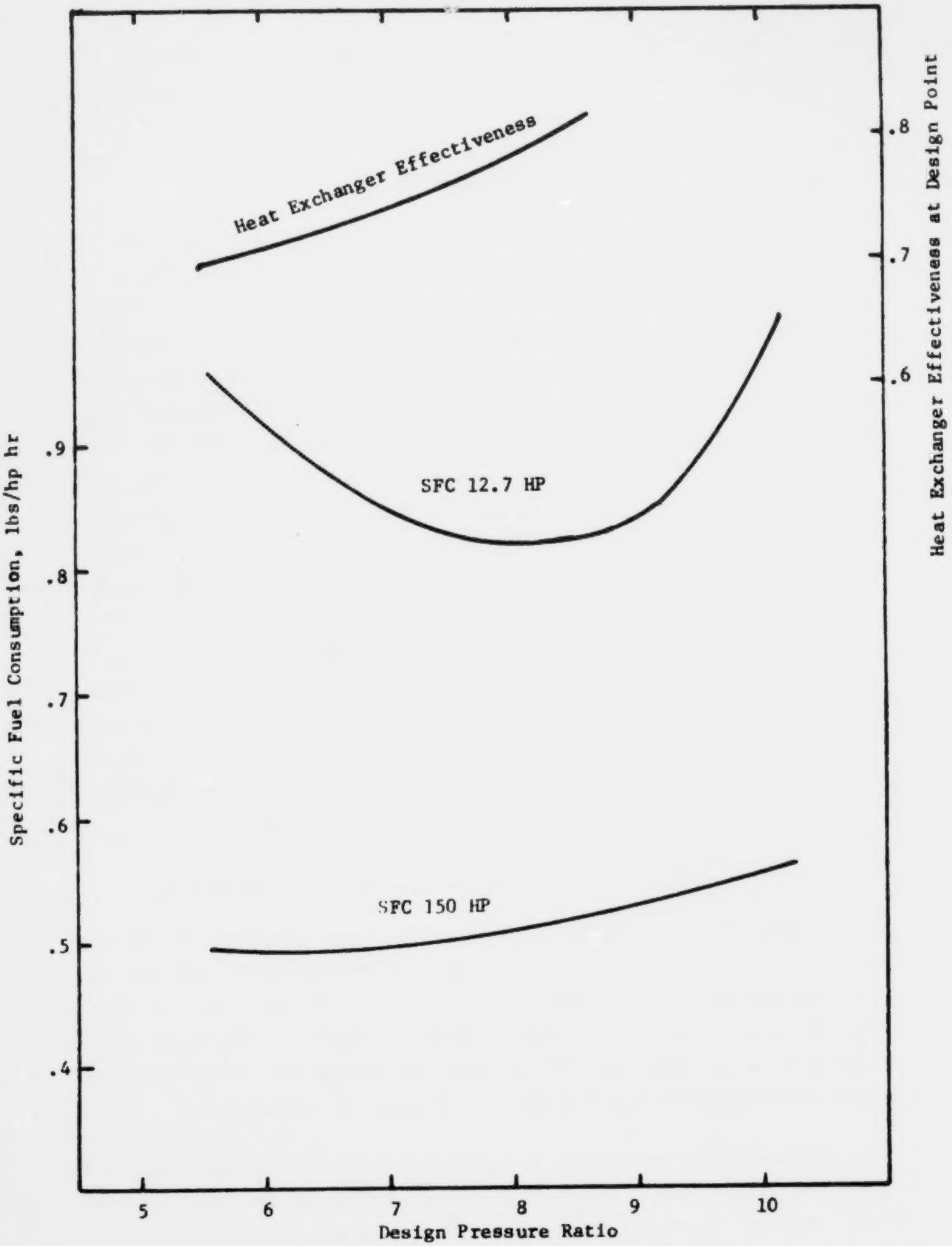


Figure 48. Specific Fuel Consumption and Heat Exchanger Effectiveness vs. Design Pressure Ratio, PD-2 Engine

1. Reduction in size, weight and cost of main engine housing. This is particularly important in view of the high pressure and temperature of the compressor discharge air.
2. Elimination of rotating regenerator seals.
3. Elimination of rotating regenerator drive train.

Accordingly the engine physical configuration shown in Figure 49 was selected. This is a free-turbine engine configuration with two recuperators. One recuperator is mounted on either side of the cylindrical engine housing. The compressor inlet is at the front and the reduction gears are at the rear. Accessories are driven from the power turbine. The starter is coupled to the gas generator through a bevel gear drive. The combustor is mounted at the top of the engine and discharges radially into the gas generator turbine scroll. A detailed description of this configuration is given below.

3.4.2.2.3 Final Selection of Speed and Pressure Ratio

The final selection of the gas generator design-point speed and engine design-point pressure ratio was governed by the acceleration distance in ten seconds requirement. In order to meet this requirement at a design point power level competitive with that of engine PD-1 (134 HP) it was found to be necessary to limit the gas generator rotor kinetic energy to a value of $.95 \times 10^6$ in-lb. At the same time stress limits at the root of the gas generator turbine buckets (IN 713LC material) for 1900°F turbine inlet temperature require a speed of no greater than 80,000 RPM. It was found that the value of $.95 \times 10^6$ in-lb kinetic energy could be achieved at a speed of 80,000 RPM with a two stage (axial-centrifugal) aluminum (335) alloy compressor having a pressure ratio of 6.6. The use of higher pressure ratios at this turbine-stress-limited speed of 80,000 RPM required compressor and turbine rotor diameters too great for maintenance of the $.95 \times 10^6$ in-lb gas generator rotor kinetic energy value. It was therefore decided to select the value of 80,000 RPM gas generator speed and 6.6 pressure ratio for the PD-2A engine. For the power turbine speed the maximum value of 50,000 RPM was selected on the basis that this is the highest speed permitting the use of class 5 ball bearings of the required size. The resultant bearing load due to a gear load combined with a thrust load, on this shaft is sufficiently low that a 20 mm bearing running

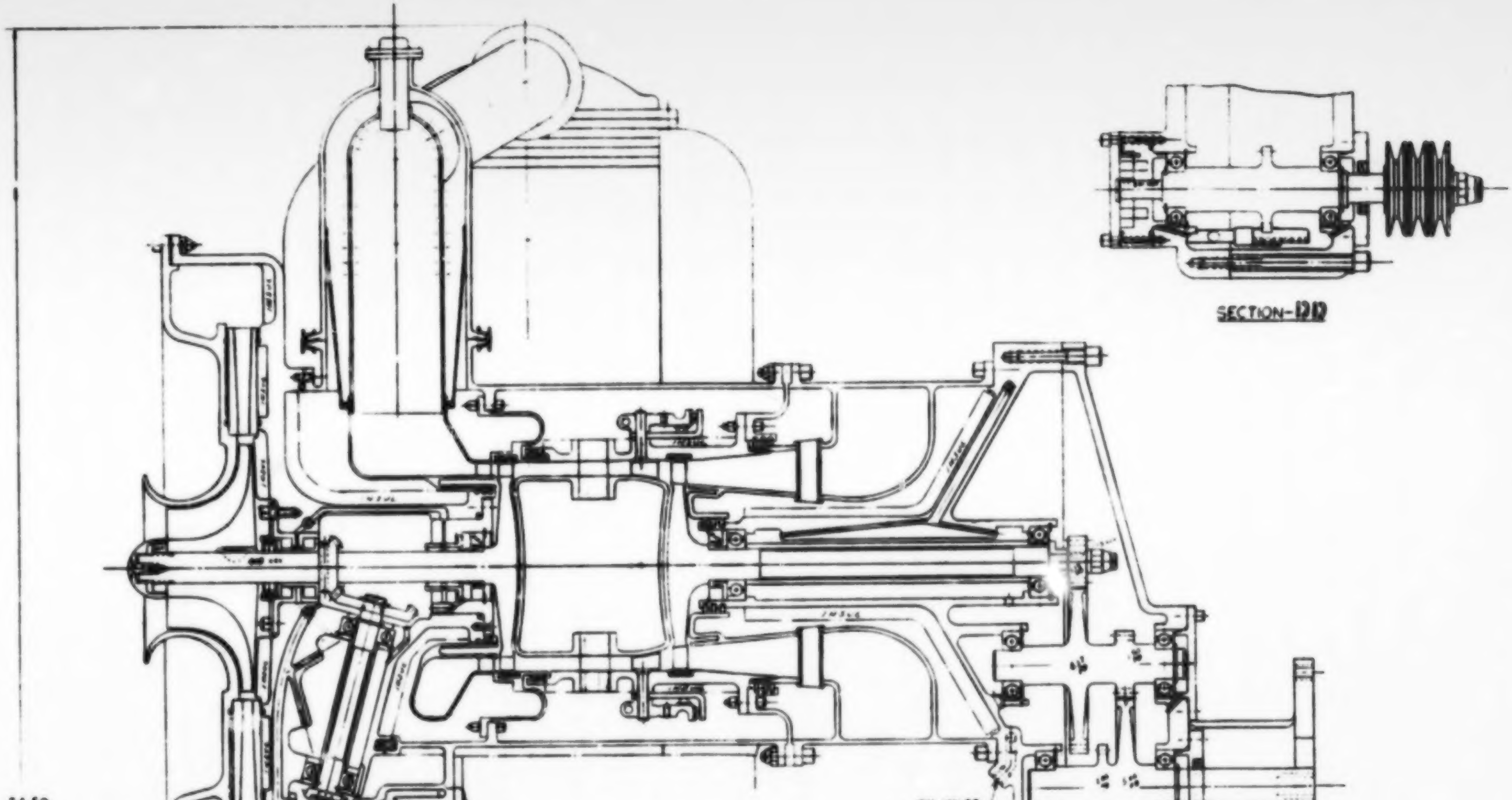

24.50
STARTER
OIL INLET - NOT IN TRUE POSITION SEE VIEW BB
SECTION-B-B
SECTION-A-A

Figure 49. Free-Turbine Engine with 1000°F Combustor

at 50,000 RPM can achieve a 1000 hour B_{10} life.

3.4.2.2.4 Description of Engine Design

General - Layout drawings of the PD-2 engine are shown in Figures 50 , 51 , 52 , and 53 . These drawings were finalized prior to the completion of the acceleration analysis. They correspond to the originally selected value of 8 to 1 pressure ratio and to the employment of a single stage steel rotor compressor running at 72,000 RPM. In these respects the drawings do not conform to the final analytical determination of engine specifications. However, the basic features of the engine are not affected by these changes.

The engine is a free-turbine configuration with variable power-turbine nozzle vanes. Two cross flow recuperators are mounted at the sides of the engine. The engine is designed for installation in the vehicle with the compressor at the front. Overall dimensions of the engine are 24.5 inches high x 27 inches wide x 31.5 inches long. The output shaft is at the rear. Accessories are driven off the power turbine, with only the starter coupled to the gas generator shaft through a bevel gear drive. The airflow enters the compressor at the front of the engine, and passes through the stationary pipe diffuser to the compressor discharge scroll. From the bottom of this scroll two symmetrically located flexible ducts lead to the bottom inlet plenums of the two recuperators. From the top outlet plenums of the recuperators ducts lead to the combustor. Hot gas from the combustor discharge enters the gas generator inlet scroll which is integral with the turbine nozzle. From the gas generator turbine the flow passes through the annular inter-shaft transition flow duct formed by the power turbine shroud-nozzle support structure and a strut supported inner body. After passing through the power turbine (variable) nozzle and wheel the flow enters the power turbine diffuser. From this diffuser the flow is directed through symmetrically positioned outlet ducts into the aft side plenums of the recuperators. The turbine exhaust passes in a forward direction through the low-pressure passages of the cross flow recuperator cores into the forward side plenums of the recuperators and then into the attached exhaust ducts in which the flow is redirected rearward to the exhaust outlets at the rear of the vehicle.

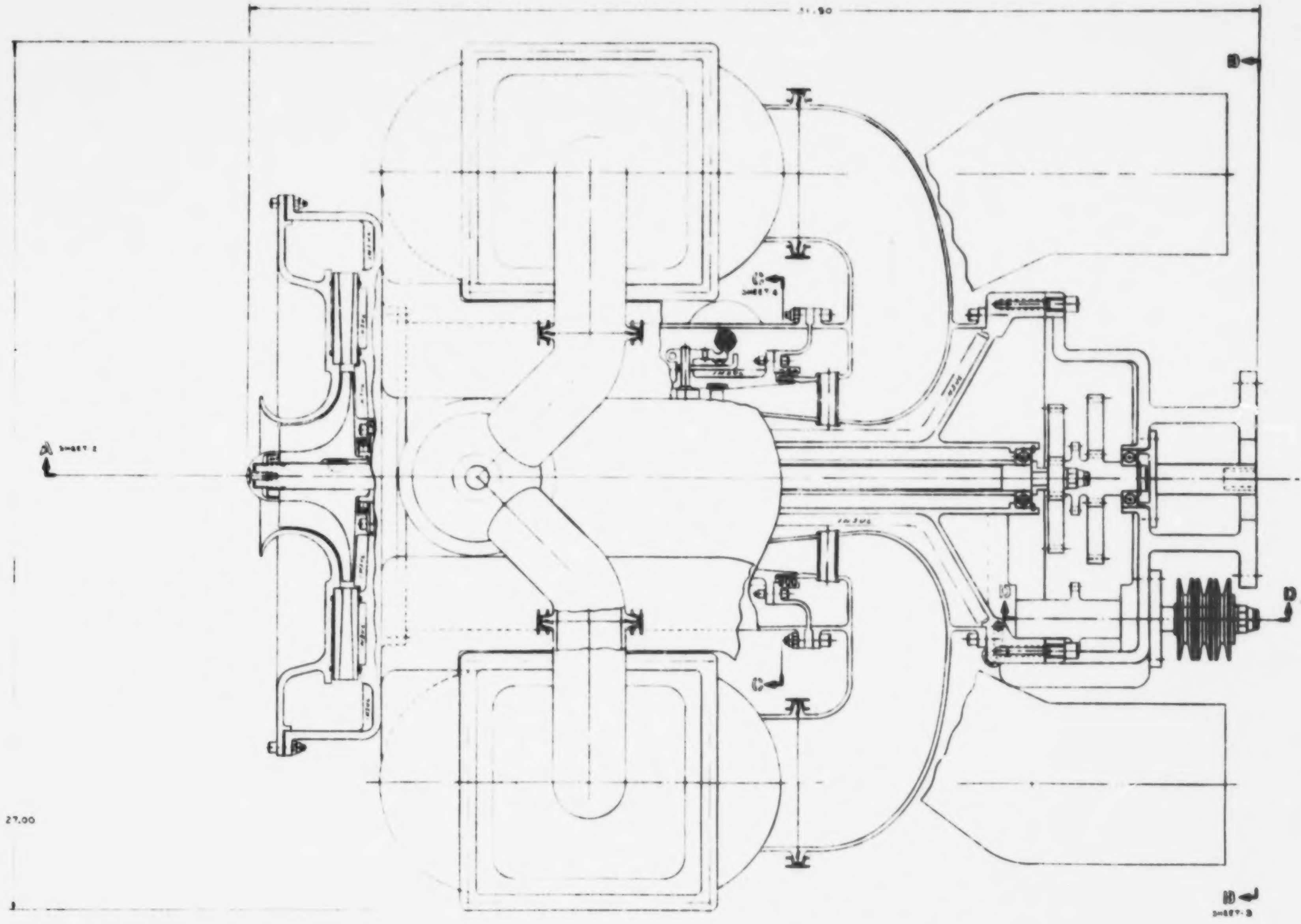

A
A
B
B
C
C
D
27.00

Figure 50. PD-2 Engine Drawing. Plan View.

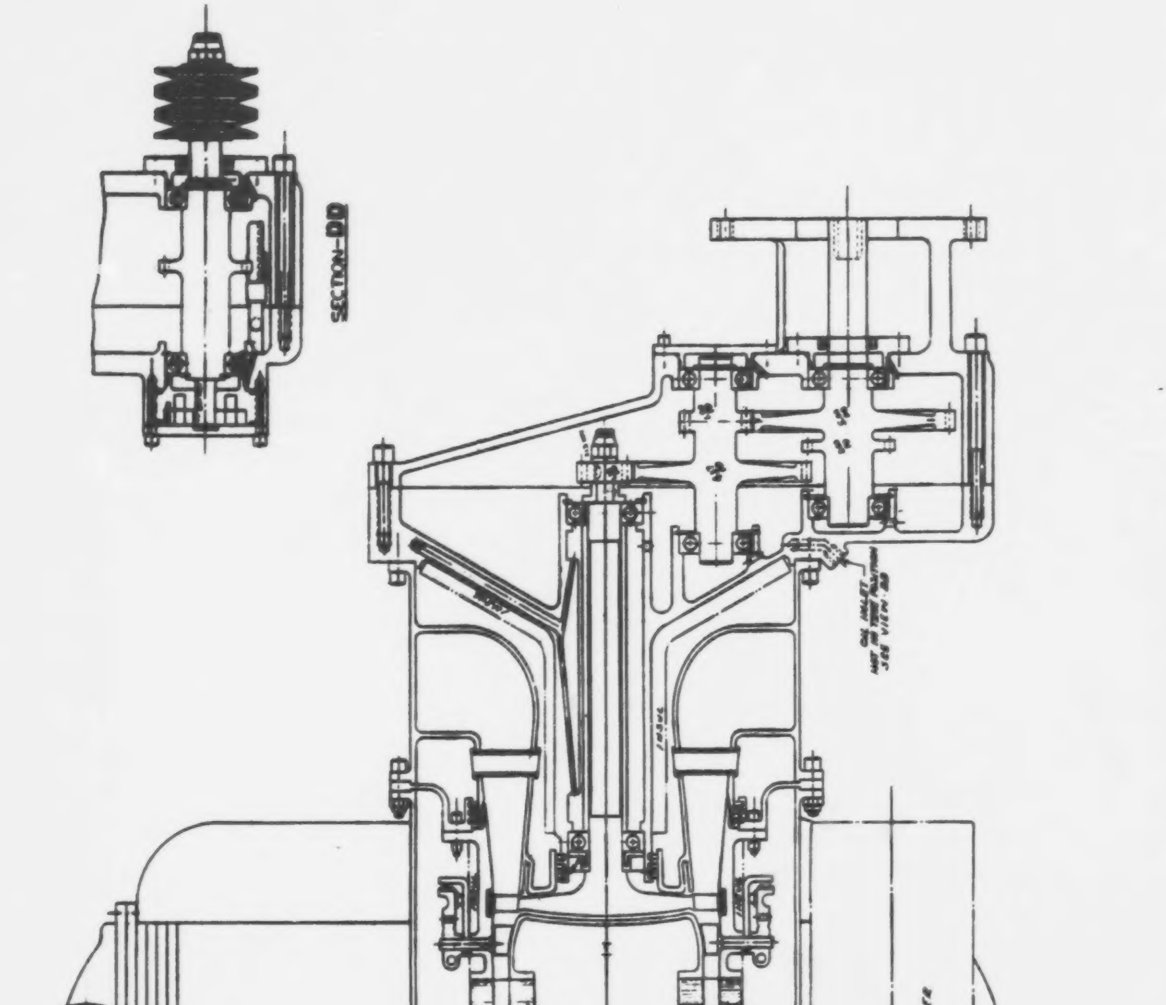

SECTION-DD
SECTION-AA
STARTER
OIL INLET
24.50

Figure 51. Free-Turbine Engine With 1000°F Combustor.

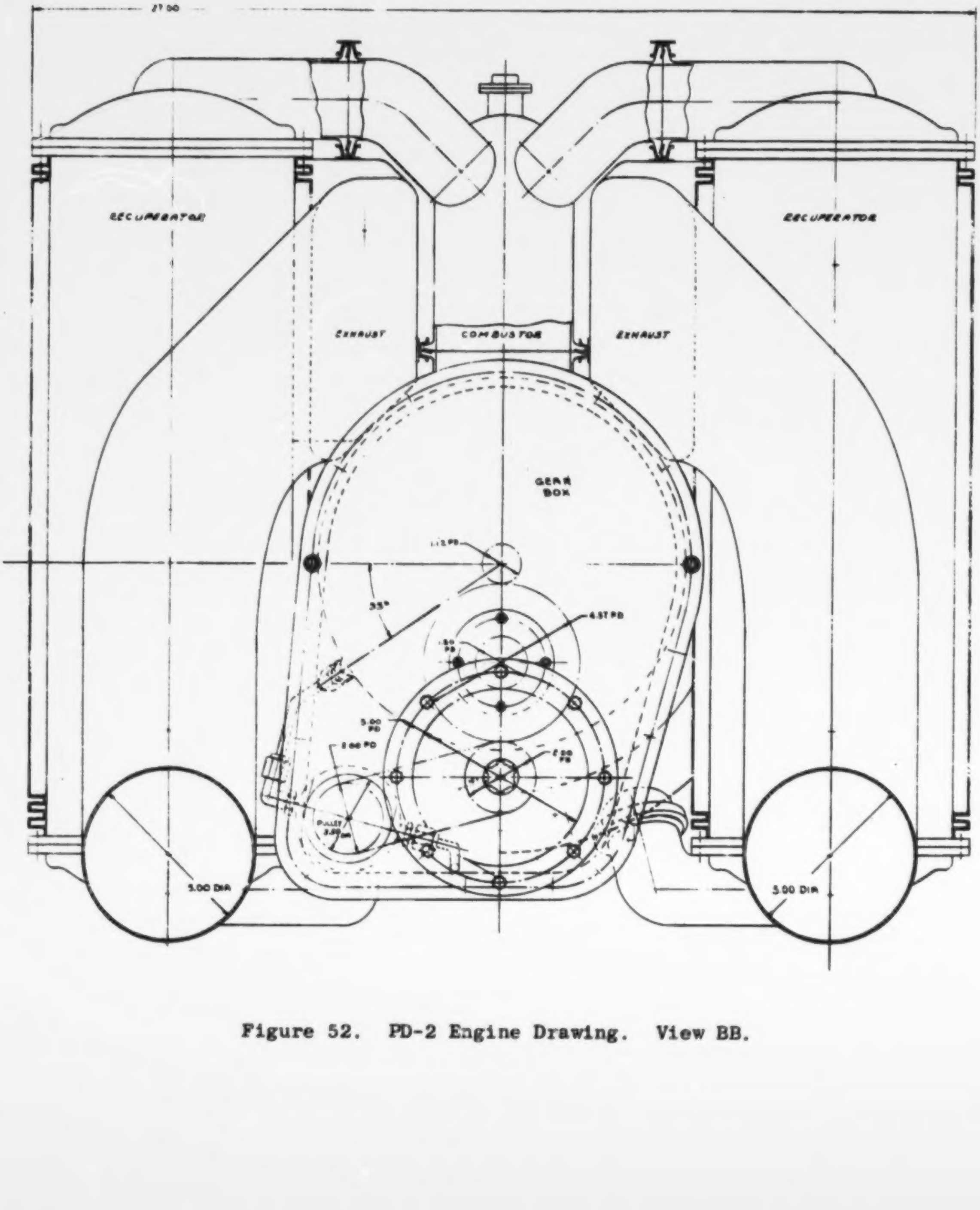

27.00
RECUPERATOR
RECUPERATOR
EXHAUST
COMBUSTOR
EXHAUST
GEAR BOX
1.12 PD
35°
4.57 PD
1.50 PD
5.00 PD
2.00 PD
PULLEY
5.00 DIA
5.00 DIA

Figure 52. PD-2 Engine Drawing. View BB.

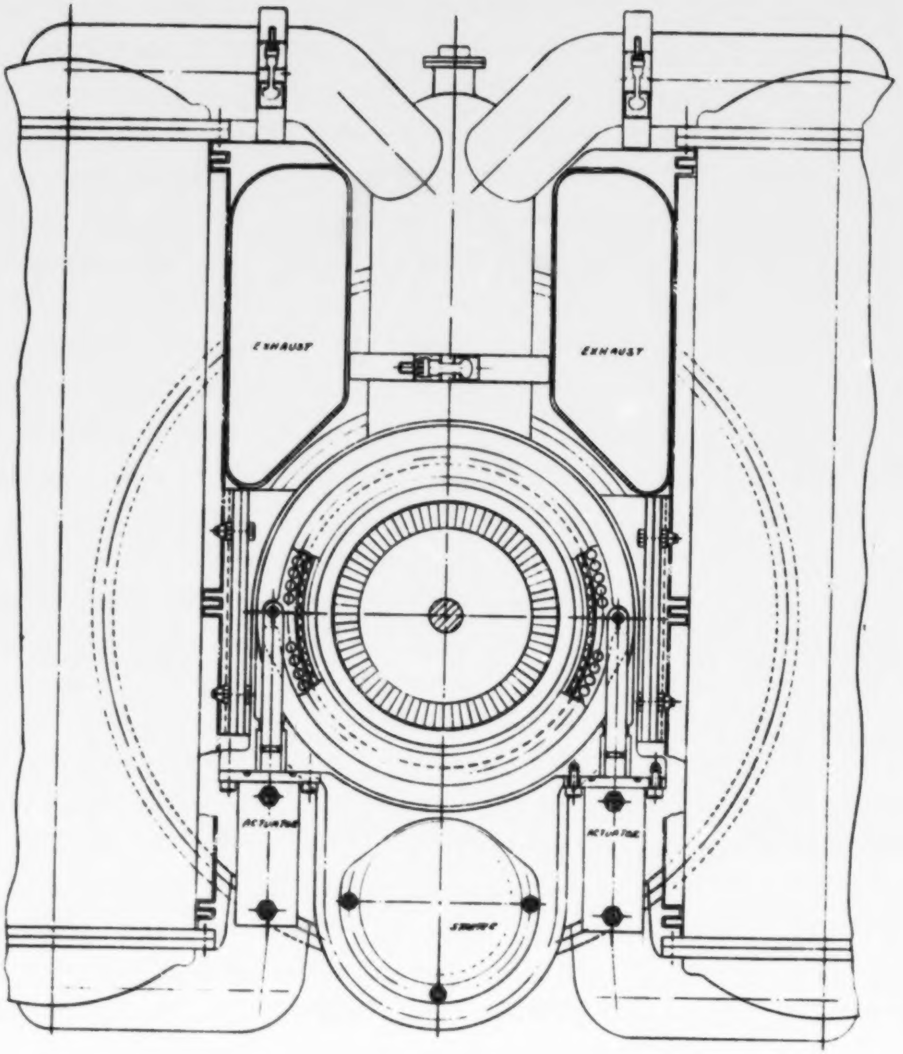

EXHAUST
EXHAUST
ACTUATOR
ACTUATOR
STARTER

Figure 53. PD-2 Engine Drawing. Section CC.

The principal structural parts of the engine rotating unit are the gas generator bearing support housing (nodular iron) which is integral with the compressor rear shroud-scroll, the forward engine casing (cast steel) which incorporates the combustor mounting flange, the rear engine casing, (304 SS) which incorporates the power turbine diffuser and turbine exhaust flanges, the power turbine bearing support housing which includes the forward section of the reduction gear casing, and the rear reduction gear casing, to which the transmission is bolted. The gas generator bearing support housing includes bearing supports for the starter drive bevel gear shaft. The starter is bolted to a flange on the forward engine casing, and incorporates an overrunning clutch which prevents torque reversal after startup. The lower part of the forward engine casing forms the gas generator oil sump, in which is mounted the oil pump supplying lubricant to the gas generator shaft bearings and the starter drive bearings. The power turbine reduction gear, and accessory drive bearings are lubricated by oil from a pump located in the rear oil sump at the bottom of the gear casing.

The gas generator turbine shroud structure (Hastelloy X) which is a part of the scroll and nozzle assembly is bolted to the forward engine housing. The power turbine shroud structure (304 SS) which includes supports for the variable vanes and for the actuating ring gear is bolted to a mounting ring secured between the forward and rear engine housings. Piston ring seals are employed between these two structures and also between the turbine diffuser and the power turbine shroud structure. An important feature of the design is that the space between all hot gas flow path structures and the outer casing is filled with compressor discharge air. This aids in the control of the nozzle actuator linkage temperature. Insulation may be applied to the inside of the outer casing to control the casing temperature to a level well below compressor discharge temperature.

The gas generator shaft is mounted on dual-film slipper type bearings manufactured by the Clevite Corporation. These bearings are an inherently stable type of high speed sleeve bearing. Also, because the dual film reduces the individual film rate of shear, the power loss is lower than for conventional sleeve bearings. The bearing aft of the compressor is a combination radial and thrust bearing. (The slippers carry load on two surfaces at right angles.) The power turbine shaft and all drive gear shafts are mounted on class 5 ball bearings. The use of sleeve bearings for the gas generator shaft results in full power bearing losses of 11.1 HP, compared with 1.2 HP for ball bearings. Because of the high gas generator speed required by the acceleration specifications, sleeve bearings were felt to be mandatory, since aircraft quality ball bearings capable of achieving adequate life at 80,000 RPM cost about $75, compared with about $3 for the slipper type bearings. The turbine rotor material is IN 713LC for the gas generator and CRM-6D for the power turbine.

The variable power turbine nozzle vanes (silicon nitride) are actuated by individual vane gear segments which mesh with a ring gear that is rotated by two hydraulic actuators mounted on the outside of the forward engine casing. Approximately 120° of nozzle rotation are provided for in order to accommodate dynamic braking of the engine.

The recuperators consist of ceramic cores made up of alternate layers of small tubes oriented at right angles, supported inside steel shells. The high pressure flow path is sealed by use of spring loaded seals at the square flanges between the upper and lower plenum covers and the ceramic core. The low pressure flow path is sealed by spring leaves between the housing and the core. The housing is reinforced on its flat side to withstand the pressure of the low pressure flow path.

Engine Installation - In Figures 54, 55, and 56 the PD-2 engine is shown installed in a standard six-passenger automobile. These figures show that the space available is more than large enough to accommodate the engine.

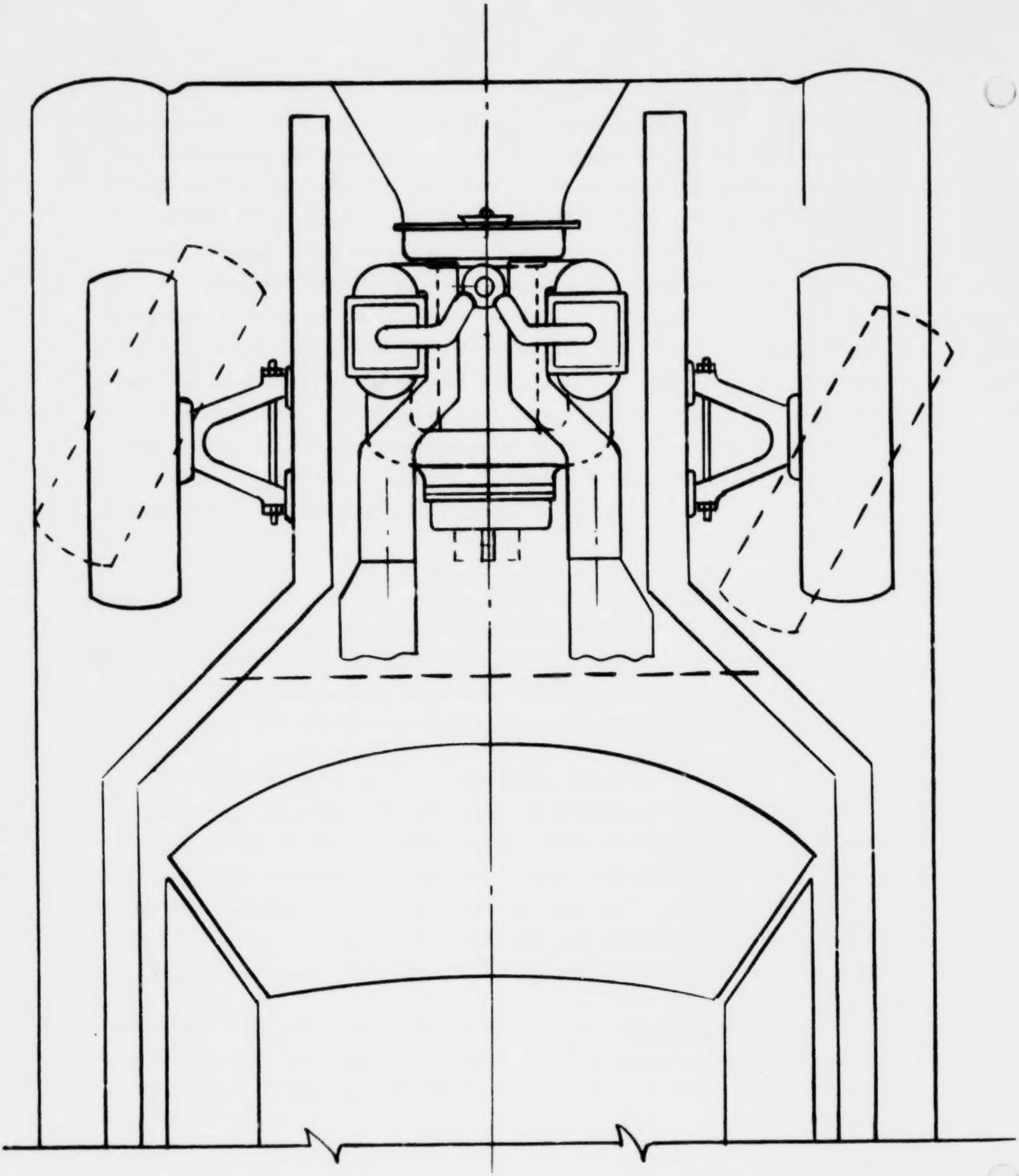


Figure 54 PD-2 Engine Installed in a Standard Six-Passenger Sedan. Top View.

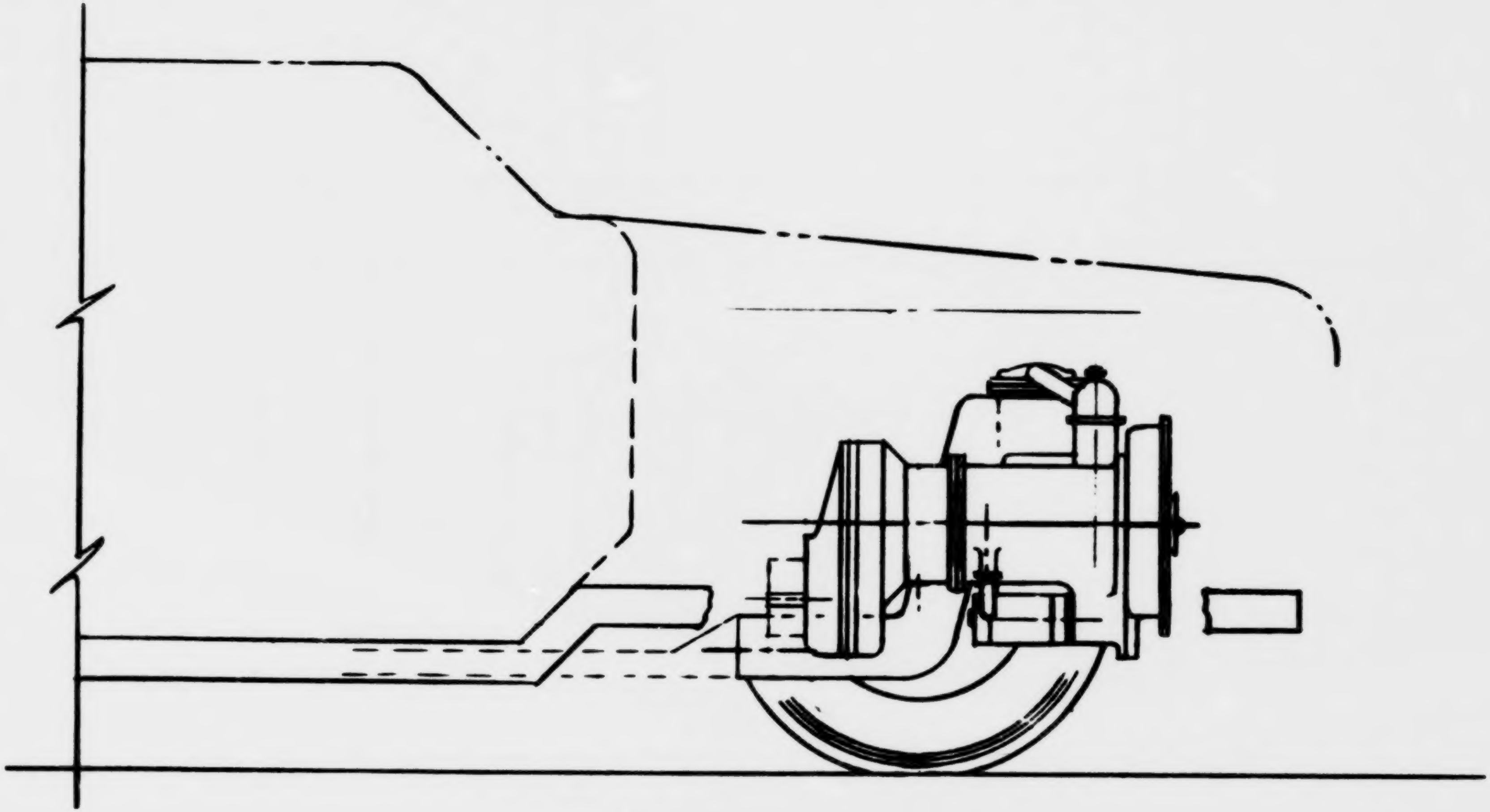


Figure 55. PD-2 Engine Installed in a Standard Six-Passenger Sedan. Side View.

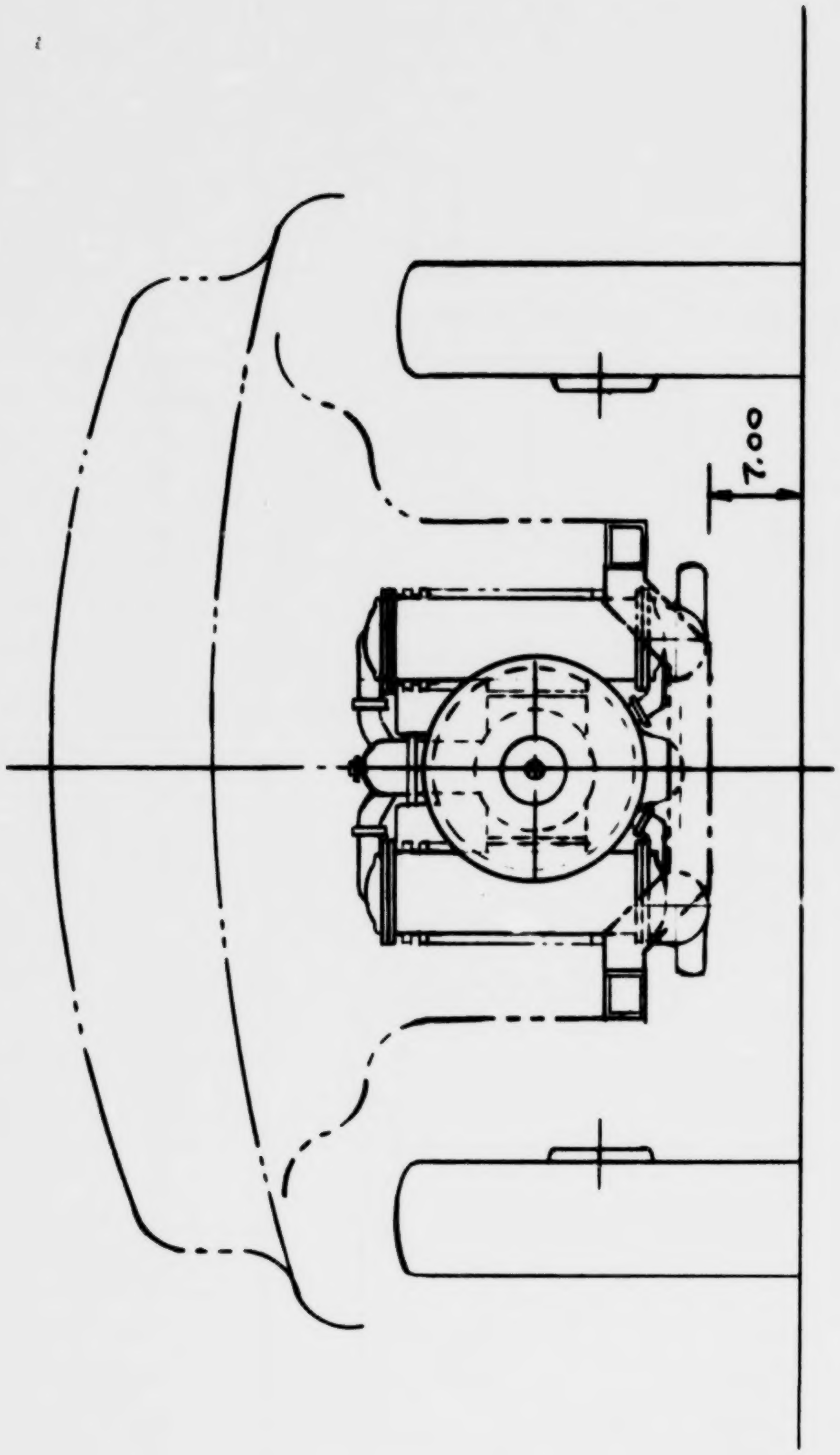


Figure 56. PD-2 Engine Installed in a Standard Six-Passenger Sedan. Front View.

3.5 Off Design Performance

3.5.1 Code Description

A General Electric computer code for a two spool,shaft power gas turbine engine was adapted to the automotive gas turbine application. For a free power turbine configuration the low pressure compressor subroutine is bypassed, and for a single shaft engine the high pressure compressor and turbine are bypassed. The compressor maps shown in Figures 1 and 2 were used for the low pressure and high pressure compressor respectively.

A subroutine was written for the regenerator using the model described in Section 3.1.3. This subroutine was subsequently modified for the PD-2 engine to incorporate the effectiveness - NTU relationship of the ceramic cross-flow recuperator.

The turbine maps were incorporated into the program using a modeling technique developed in GE which reduces the turbine maps for any nozzle area to essentially two single line plots, one for flow and one for efficiency. The plots shown in Figures 5 and 6 indicate the turbine performance at 50% and 100% turbine area, but the turbine area was completely variable and was specified for performance calculations by program input.

The program is stored on a magnetic tape in the local computations center. The control data and namelist input are fed into the computer by means of a punched paper tape from a remote terminal. The first case is always a match point in which the cycle conditions are specified. These include the compressor and turbine match points on the maps, rotative speeds, power output, turbine inlet temperature, pressure drops and Mach numbers at various stations in the engine. The program will calculate the flow rate, and the flow areas required. Then for off-design calculations the flow areas are fixed and pressure drops are calculated as functions of Mach numbers.

The compressor and turbine maps are used in the off-design cycle calculations which satisfy the conservation equations of mass and energy by iteration. For off-design calculations the user can specify engine speed and power output or turbine inlet temperature. For the PD-2 engine the program was run for constant combustor inlet temperature by changing the iteration parameters. The output print out gives a complete listing of the cycle parameters, including pressures, temperatures, flow rates, speeds, power and fuel consumption.

3.5.2 Design Point Values

The inputs to the cycle calculation code are listed in Table 14 for the four engines that were evaluated. Engines CD-1 and PD-1 are regenerative, single shaft engines with variable geometry in the compressor and turbine. Engines CD-2 and PD-2 are two shaft engines with variable geometry in the power turbine only. Engine CD-2 has a rotating regenerator and PD-2 has a stationary recuperator.

All engine performance was calculated for ambient conditions of 14.7 psia and 105°F. The inlet pressure to the engine was reduced one percent for losses in the inlet and filter. The air flow rates shown were calculated for 150 HP engine output. The compressor efficiency for engines CD-1 and PD-1 were reduced about three percent because of their variable geometry. The cooling air was assumed to be three percent for the first two engines, based on a preliminary analysis, and raised to five percent for the latter two engines. The heat exchangers were designed for 8 percent pressure drop with most of it on the hot side. The cold side pressure drop was reduced for the PD-1 engine because about one-fourth of the combustor flow bypasses the regenerator. The turbine inlet temperature was limited to 1850°F for the PD-1 engine to permit the use of low cost turbine material (CRM-6D). The accessory power was taken from the gas generator shaft for engine CD-2 and from the power turbine shaft for the PD-2 engine to reduce the equivalent inertia of the gas generator shaft. The first three engines have regenerators with design effectiveness values of 0.85, and the PD-2 engine has a recuperator. The lower effectiveness is required to limit the combustor inlet temperature to 1000°F.

3.5.3 Off-Design Performance

The off-design performance code was utilized to develop engine performance characteristics for the various engine cycles. The initial

Table 14

Inputs for Engine Off-Design Performance Calculations

ENGINE	CD-1	CD-2	PD-1	PD-2
Ambient temperature, °F	105	105	105	105
Inlet pressure, psia	14.553	14.553	14.553	14.553
Air flow rate, lb/sec	2.09	1.79	2.58	1.77
Output power, hp	150	150	150	150
Engine speeds, rpm	40K	55K/35K	32750	72K/50K
Pressure ratio	3.6	5.0	3.2	8.0
Compressor efficiency	.80	.807	.80	.787
Cooling air fraction	.03	.03	.05	.05
Leakage fraction	.04	.05	.04	.03
Cold side H.E. pressure drop, %	1	1	.75	1
Combustor pressure drop, %	4	4	4	4
Combustor efficiency	.99	.99	.99	.99
Turbine inlet temperature, °F	1900	1900	1850	1900
Turbine efficiency	.85	.84/.85	.85	.84/.85
Accessory power, hp	4	4	4	4
Turbine exit pressure drop, %	2	2	3	2
H.E. effectiveness	.85	.85	.85	.78
Hot side H.E. pressure drop, %	7	7	7	7

engine performance calculations were made for an ambient air inlet temperature of 85°F, but since the acceleration requirements of the contract are more stringent at an ambient air temperature of 105°F, all subsequent calculations were made for an inlet temperature of 105°F. The engines were sized for 150 HP output using the inputs listed in Table 14. Then for off design performance calculations the turbine inlet temperature was reduced, holding fixed geometry, to 1700°F which was considered the maximum turbine inlet temperature for continuous operation. Then for reduced power the variable geometry was used to maintain the turbine inlet temperature at 1700°F, or as high as possible. For very low power levels the turbine inlet temperature was reduced.

For the free power turbine engines the operating parameters were shaft power output, power turbine speed, and power turbine nozzle vane effective area. Parametric variations of these parameters were made to develop a complete engine performance map. Limiting parameters such as turbine inlet temperature and compressor stall margin determined the bounds of the engine operating map.

Correlations of engine parameters were developed by plotting turbine inlet temperature versus shaft power output for each turbine nozzle vane setting, with power turbine speed as a parameter. For constant turbine inlet temperature, plots of power output and fuel flow rate were made as functions of power turbine speed and vane setting.

The objective of these calculations was to determine the minimum fuel flow rate for a given power output and engine speed. The best specific fuel consumption (fuel flow rate/shaft power output) generally occurs when the turbine inlet temperature is at its maximum allowable value, limited by compressor stall margin requirements. The purpose of the optimization was to determine the schedule of variable geometry settings that would give the best engine performance over the operating range. The final result of the calculations was an engine performance map of power output versus power turbine speed with fuel flow rate as a parameter, shown in Figure 57 for the CD-2 engine. This engine map was used for mission analysis calculations. Shown in Figure 58 is the schedule used for the power turbine nozzle vanes in calculating the engine performance. Initially the design point was matched with the power turbine vanes at

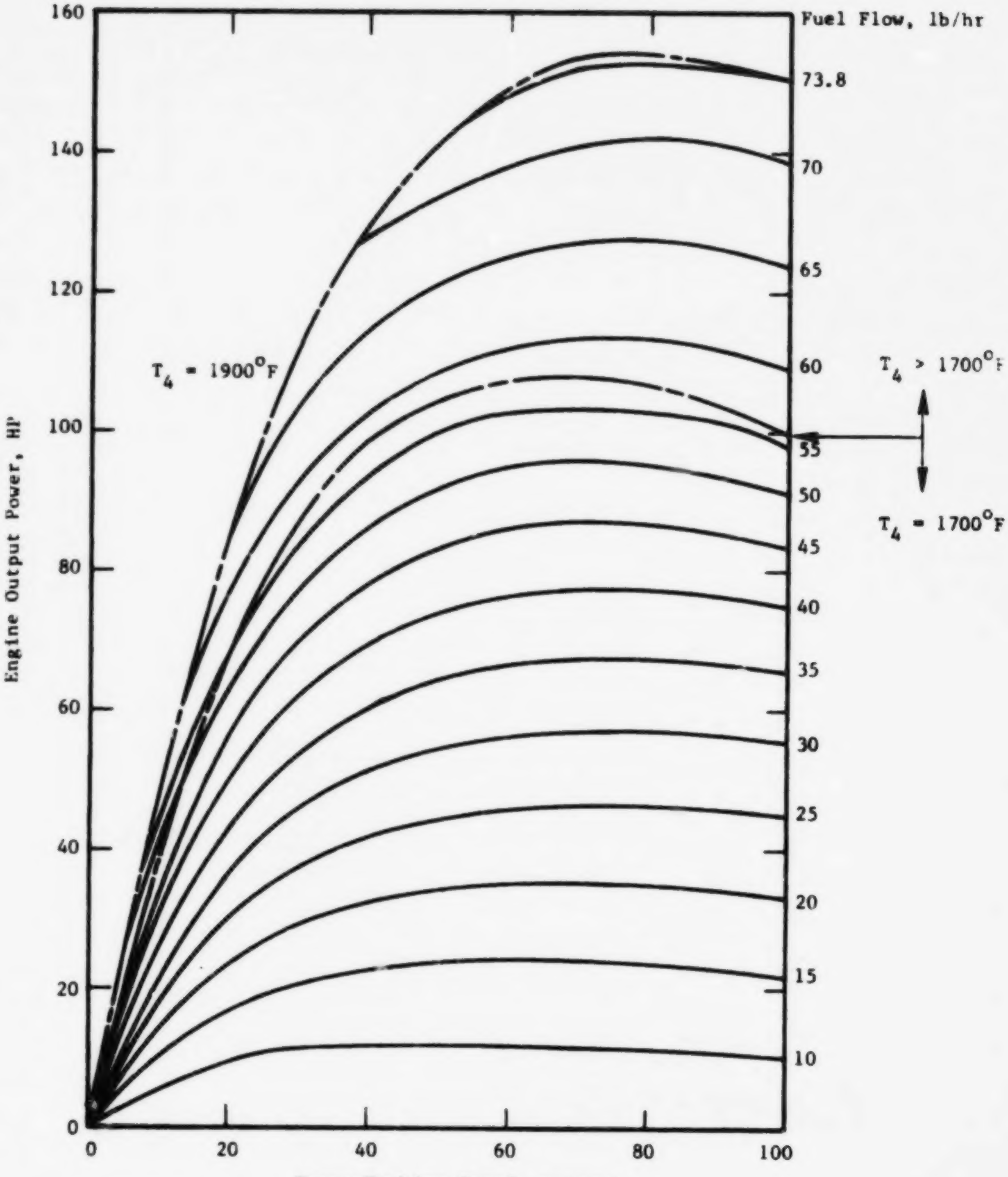


Figure 57. CD-2 Engine Performance, 105°F Ambient Temperature.

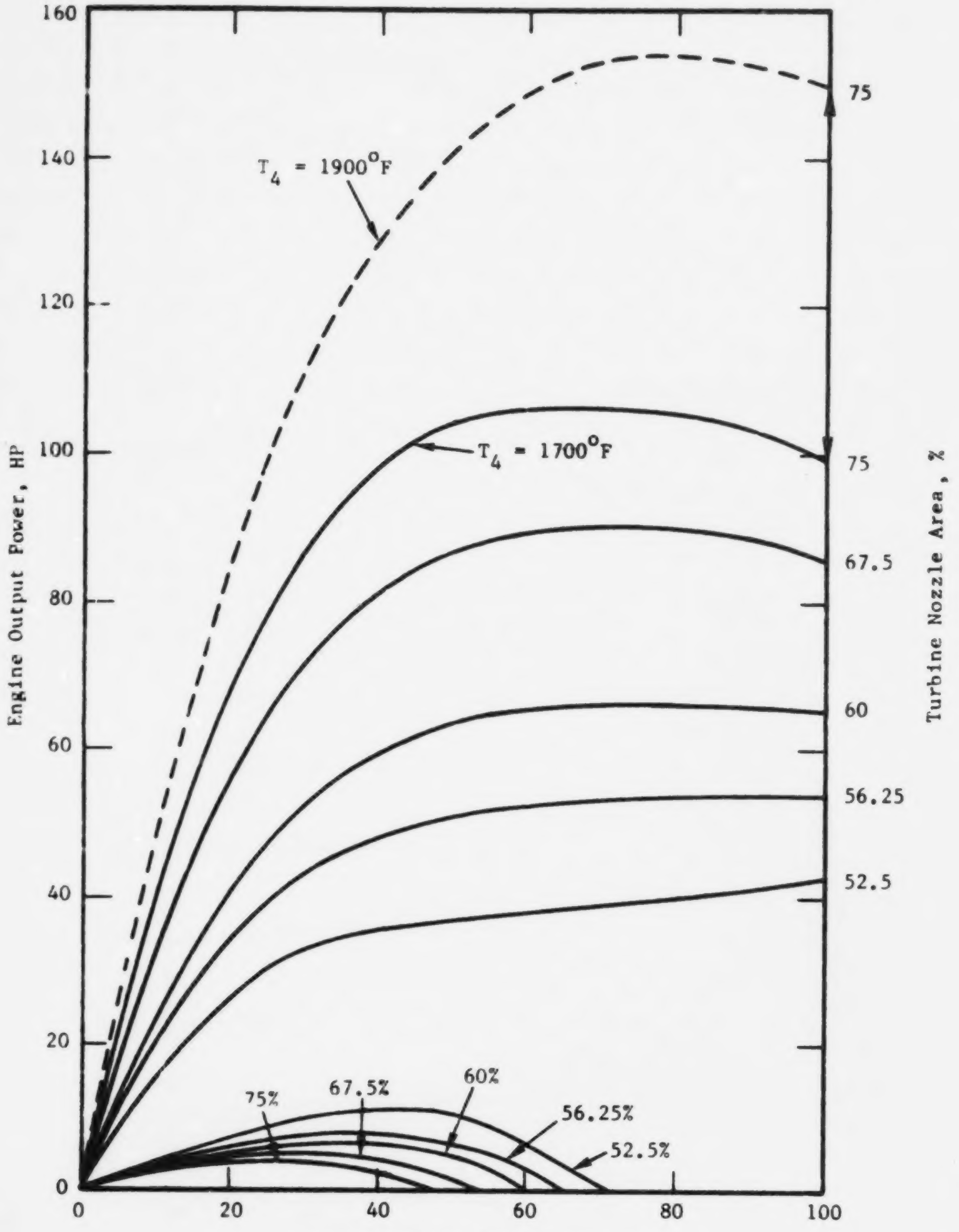


Figure 58. CD-2 Engine Power Turbine Nozzle Areas.

100 percent area and varied down to 70 percent for off design calculations. Then the transient analysis indicated the desirability of greater than design nozzle area for acceleration, and the engine was rematched at 75% turbine nozzle area.

Similar performance calculations were made for the PD-2 engine except that the combustor inlet temperature was limited to 1000°F to reduce the NO_x emissions. A change in the cycle iteration parameters permitted operation at constant combustor inlet temperature, and engine performance was calculated for variations in power turbine speed and nozzle vane setting. Since the turbine inlet temperatures never exceeded the maximum allowable value of 1900°F, the correlations were simpler than for other free turbine engines. At low power and low speeds the combustor inlet temperature was less than 1000°F. The engine performance map of engine power versus power turbine speed with fuel flow rate as a parameter, shown in Figure 59 . On this map the turbine inlet temperature was allowed to vary, but combustor inlet temperature was maintained at 1000°F. The line of 1700°F turbine inlet temperature is shown as it may be of interest. The turbine inlet temperature is higher for power levels above the line and lower for those below the line. The schedule of power turbine nozzle area used for these calculations is shown in Figure 60. The power turbine speed was not allowed to go below 15 percent because the accessories of this engine are driven by the power turbine.

For the single shaft engines the off-design performance maps were generated in a similar way. The engine power was first reduced from the maximum value by reducing turbine inlet temperature to 1700°F. Then at constant turbine inlet temperature the power was reduced by closing the compressor diffuser vanes and the turbine nozzle vanes simultaneously. Closing the compressor vanes reduced the air flow rate and the turbine vanes were closed to maintain as high a pressure ratio as possible with adequate stall margin. The schedule of turbine nozzle area as a function of compressor diffuser area and engine speed shown in Figure 61 was developed for the first single shaft engine evaluated and was used for all similar single shaft engines. The performance map for the CD-1 engine is shown in Figure 62 , as power output versus engine speed with fuel flow rate as a parameter. The captions at the right indicate the turbine

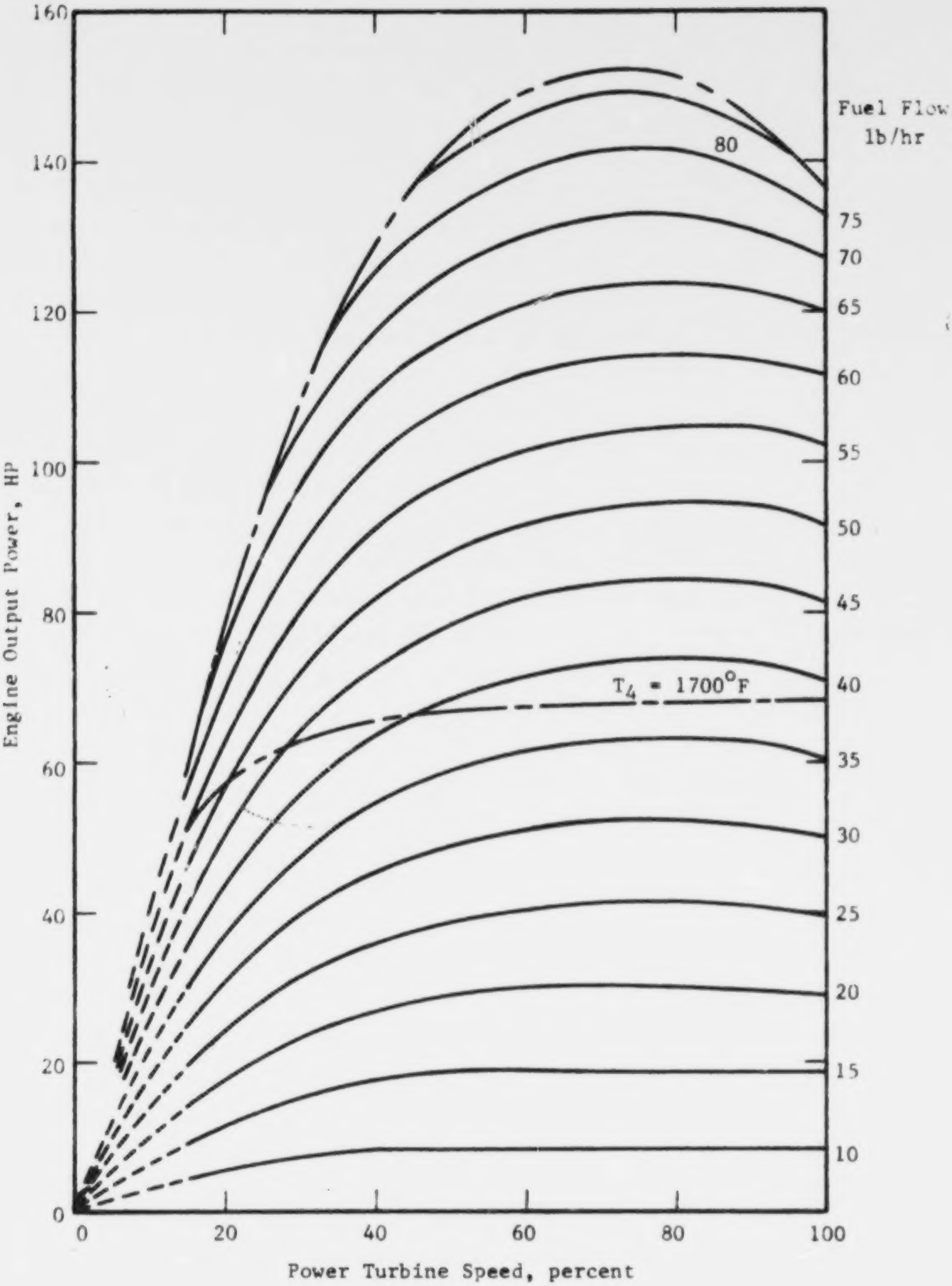


Figure 59. PD-2 Engine Performance, 105°F Ambient Temperature.

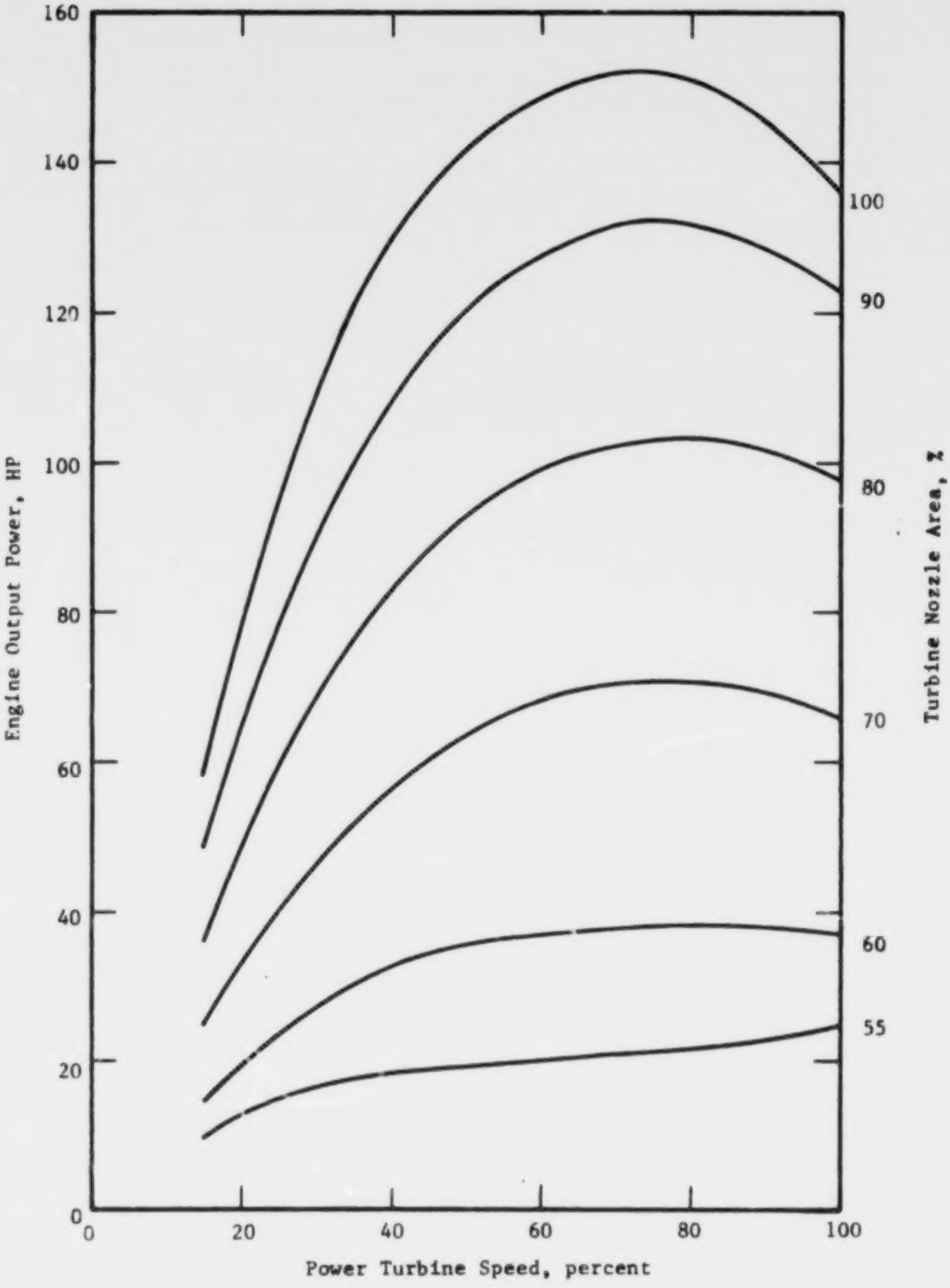


Figure 60. PD-2 Engine Power Turbine Nozzle Areas.

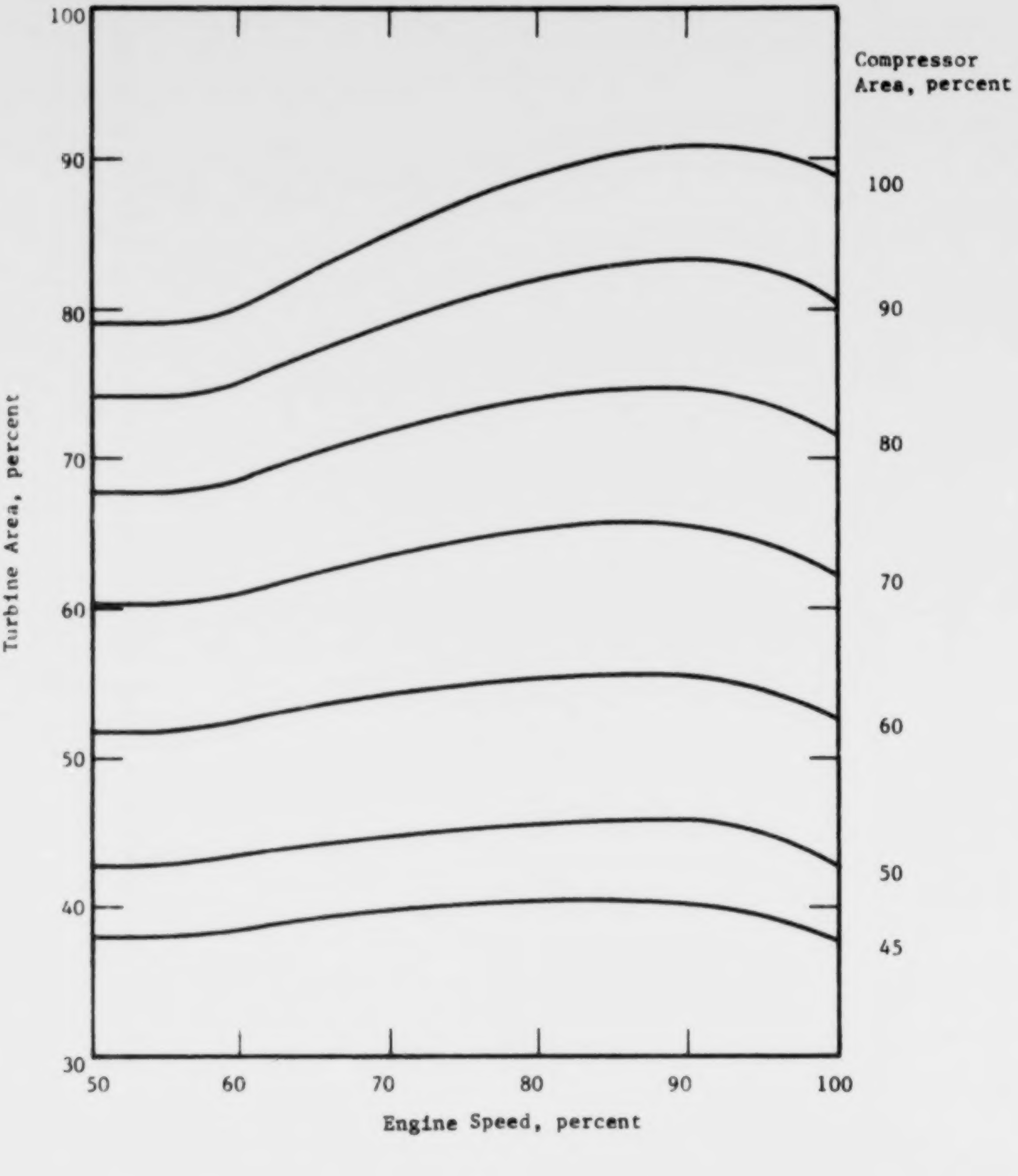


Figure 61. Turbine Area Schedule Single Shaft Engine Turbine Inlet Temperature 1700°F.

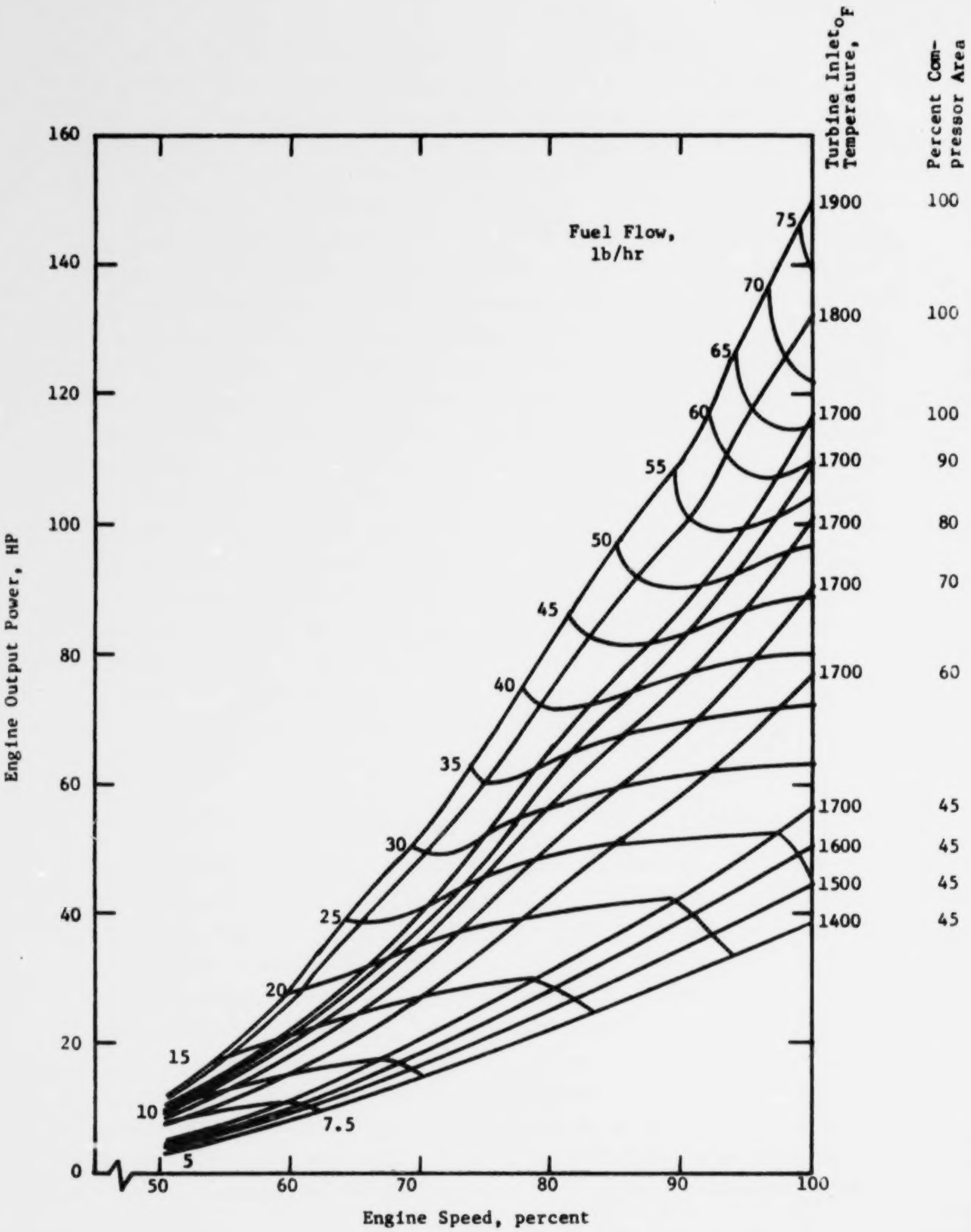


Figure 62. CD-1 Engine Performance 105°F Ambient Temperature

inlet temperature and the percent compressor diffuser area used in the calculations. A plot of fuel flow rate versus engine speed is shown in Figure 63. These maps were used for the mission analyses.

The PD-1 engine is very similar to CD-1 in that they are both regenerated, single shaft engines with low compressor pressure ratios. Therefore, the off-design performance was calculated in a similar way. The principal difference is that the PD-1 engine has a primary-air regenerator bypass system. This imposes a penalty on engine performance because it has the effect of lowering the regenerator effectiveness. A special time sharing program was written to correct the fuel flow for the bypass effect after the off-design calculations had been done in the usual way. The engine performance maps are shown in Figure 64 , as power output versus engine speed and Figure 65 , as fuel flow versus engine speed. In each figure can be seen the region of operation at 1700°F turbine inlet temperature in which the compressor and turbine geometry are varied. Above and below this region the engine is run with fixed geometry and varying turbine inlet temperature. The performance is almost the same as for engine CD-1 except that fuel flow is higher because the primary combustion air bypasses the heat exchanger. Although these maps show engine performance down to 50 percent engine speed, the idle speed was subsequently set at 70 percent to meet the acceleration requirements.

3.5.4 Modification of Off-Design Performance

After the performance calculations had been completed and the results had been reviewed it appeared desirable to modify the input assumptions for the PD-1A and PD-2A engine cycles to reflect the most recent information. A listing of the modified assumptions is shown in Table 15 along with the original values used in the performance calculations. The inlet pressure drop was reduced from one to one half percent because the air velocity at the filter is quite low. The engine speeds were raised to reduce the rotor moment of inertia for better acceleration. The compressor of the PD-2A engine was changed to a transonic axial stage followed by a radial stage and the pressure ratio (tip speed) was reduced to 6.6 to achieve a lower moment of inertia for the gas generator rotor.

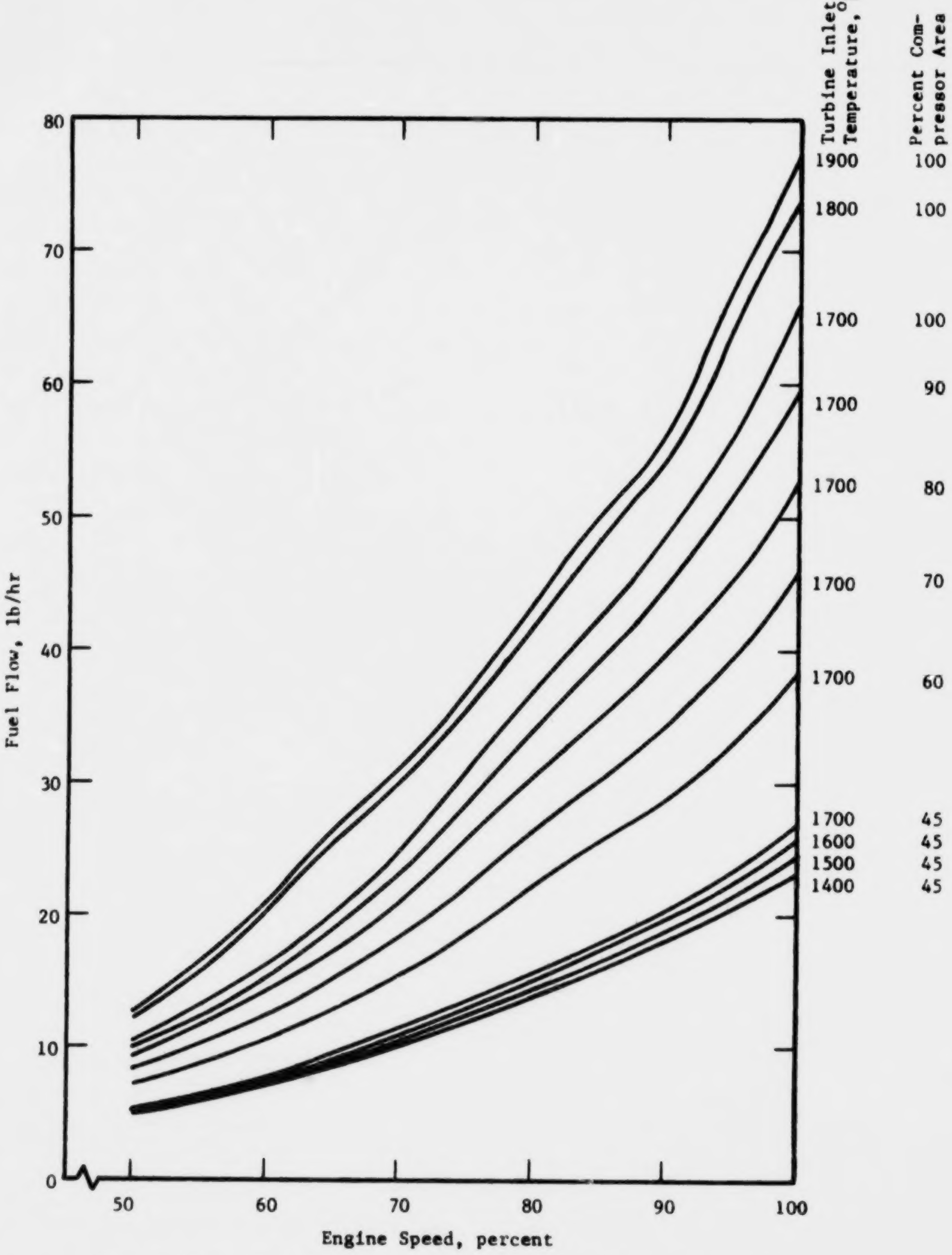


Figure 63. CD-1 Engine Performance, 105°F Ambient Temperature.

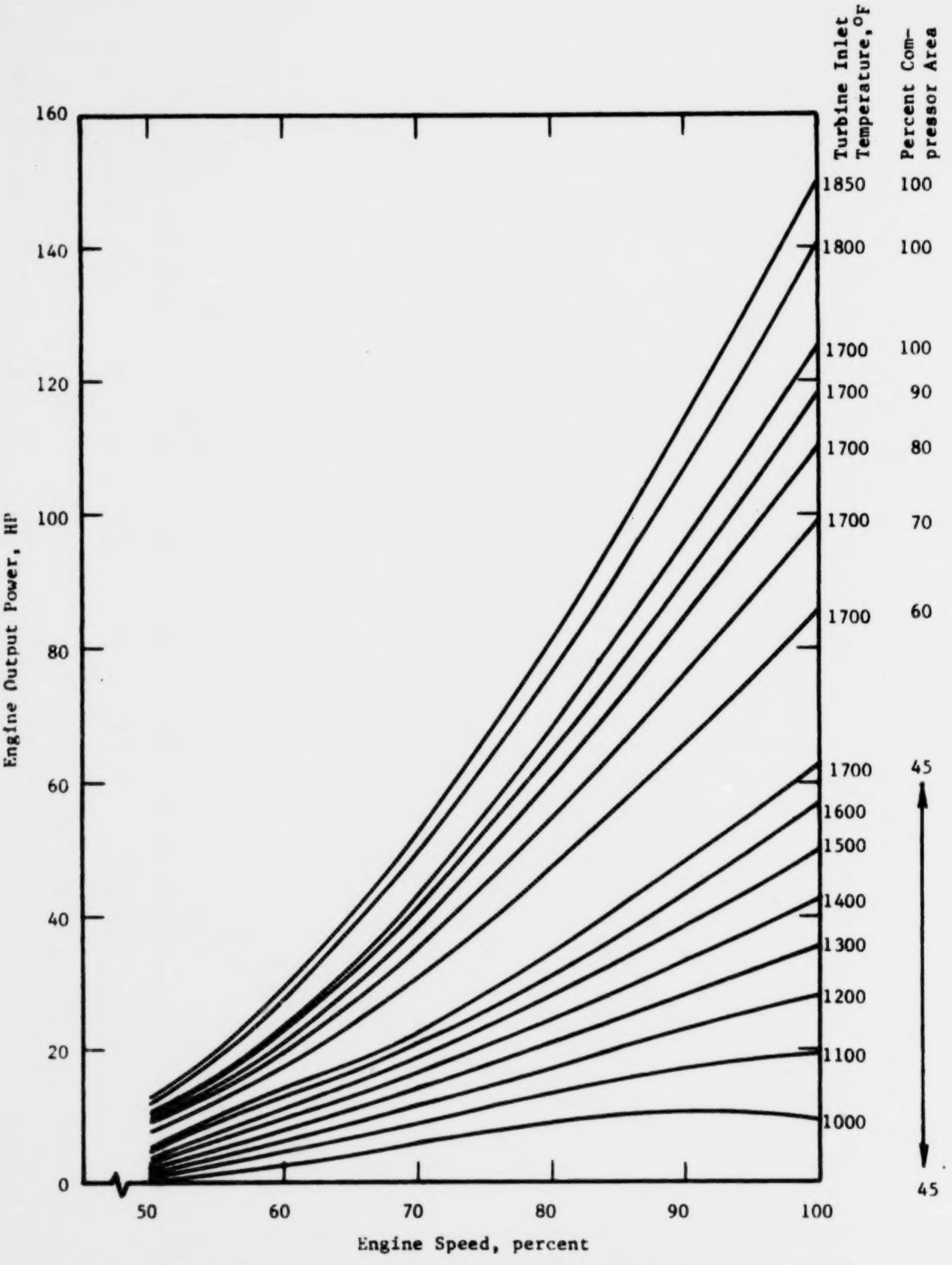


Figure 64. PD-1 Engine Performance, 105°F Ambient Temperature.

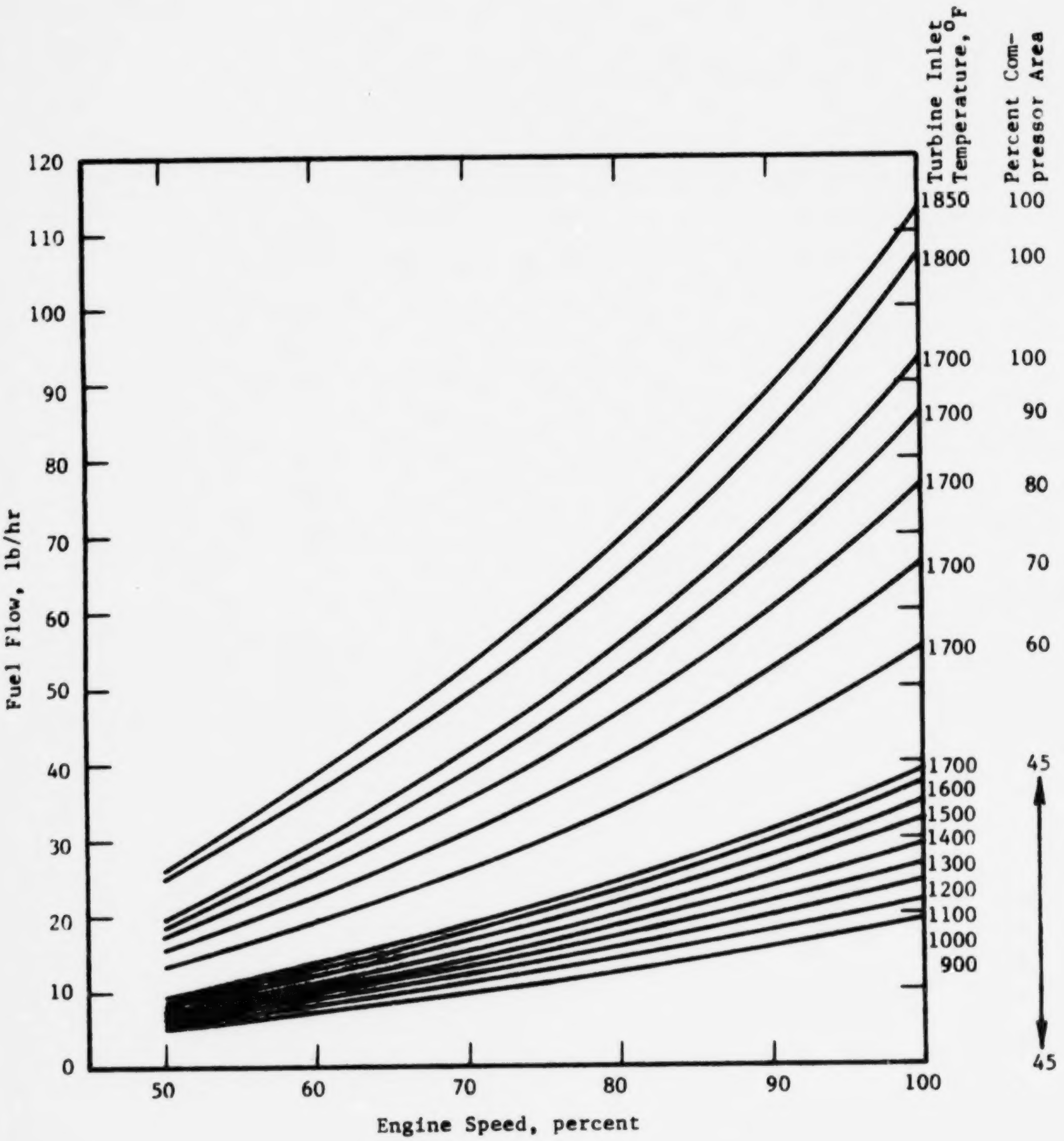


Figure 65. PD-1 Engine Performance, 105°F Ambient Temperature.

Table 15

Modified Assumptions for PD-1A and PD-2A Engines

Engine	PD-1	PD-1A	PD-2	PD-2A
Ambient temperature, °F	105	105	105	105
Inlet pressure, psia	14.553	14.626	14.553	14.626
Airflow rate, lb/sec	2.58	1.96	1.77	1.66
Output power, HP	150	150	150	150
Engine speeds, RPM	32750	40000	72K/50K	80K/50K
Pressure ratio	3.2	3.2	8.0	6.6
Compressor efficiency	.80	.823	.787	.787
Cooling air fraction	.05	.02	.05	.03
Leakage fraction	.04	.04	.03	.01
Cold side H.E. pressure drop, %	.75	.75	1	1
Combustor pressure drop, %	4	4	4	4
Combustor efficiency	.99	.99	.99	.99
Turbine inlet temperature, °F	1850	1900	1900	1900
Turbine efficiency	.85	.85	.84/.85	.84/.85
Accessory power, HP	4	4	4	4
Turbine exit pressure drop, %	3	1.5	2	1.5
H.E. effectiveness	.85	.85	.78	.73
Hot side H.E. pressure drop, %	7	4	7	7

The compressor efficiency of the PD-1A engine was raised because the value of 0.8 was considered overly conservative for a low pressure ratio compressor.

The turbine material of the PD-1A engine was changed from CRM-6D to IN 713. This permitted the use of 1900°F turbine inlet temperature for the PD-1A engine and a reduction of cooling flow to 2 percent. The cooling flow for the PD-2A engine (two turbine wheels) was set at 3 percent. The leakage was reduced for the PD-2A engine because the recuperator would have very little leakage compared with a regenerator. Reference to Table 15 indicates a decrease in rated flow for the PD-1A engine from 2.50 to 1.96 pps, thus the regenerator hot-side pressure drop reduction from 7 to 4 percent results in only slightly larger regenerator discs. The pressure drop of the PD-2A heat exchanger was not changed because the recuperator was already quite large and the rated flow changed little.

Since there was not enough time to recalculate the entire performance maps for the modified assumptions, the engine performance was calculated at the conditions of the Uniform Simplified Engine Duty Cycle, including a condition representative of the Federal Driving Cycle. The results are shown in Figure 66 for the PD-1A engine and in Figure 67 for the PD-2A engine. As can be seen in Figure 66, the higher compressor efficiency, lower cooling fraction and lower pressure drops result in significant performance improvements for the PD-1A engine. For the PD-2A engine the lower pressure ratio and lower recuperator effectiveness outweigh the smaller effects of reduced cooling and leakage for a net reduction in engine performance as seen in Figure 67. The lower recuperator effectiveness resulted from the lower compressor pressure ratio and the 1000°F limit on combustor inlet temperature.

In the Mission Analysis results to be presented, the reported fuel mileage values for the six fixed vehicle velocities are taken from the values in Figures 66 and 67 for the PD-1A and PD-2A engines, respectively. However, since the engine performance maps were not recalculated the values reported for the FDC are adjusted by the ratio of the PD-1A, -2A to the PD-1, -2 values in Figures 66 and 67.

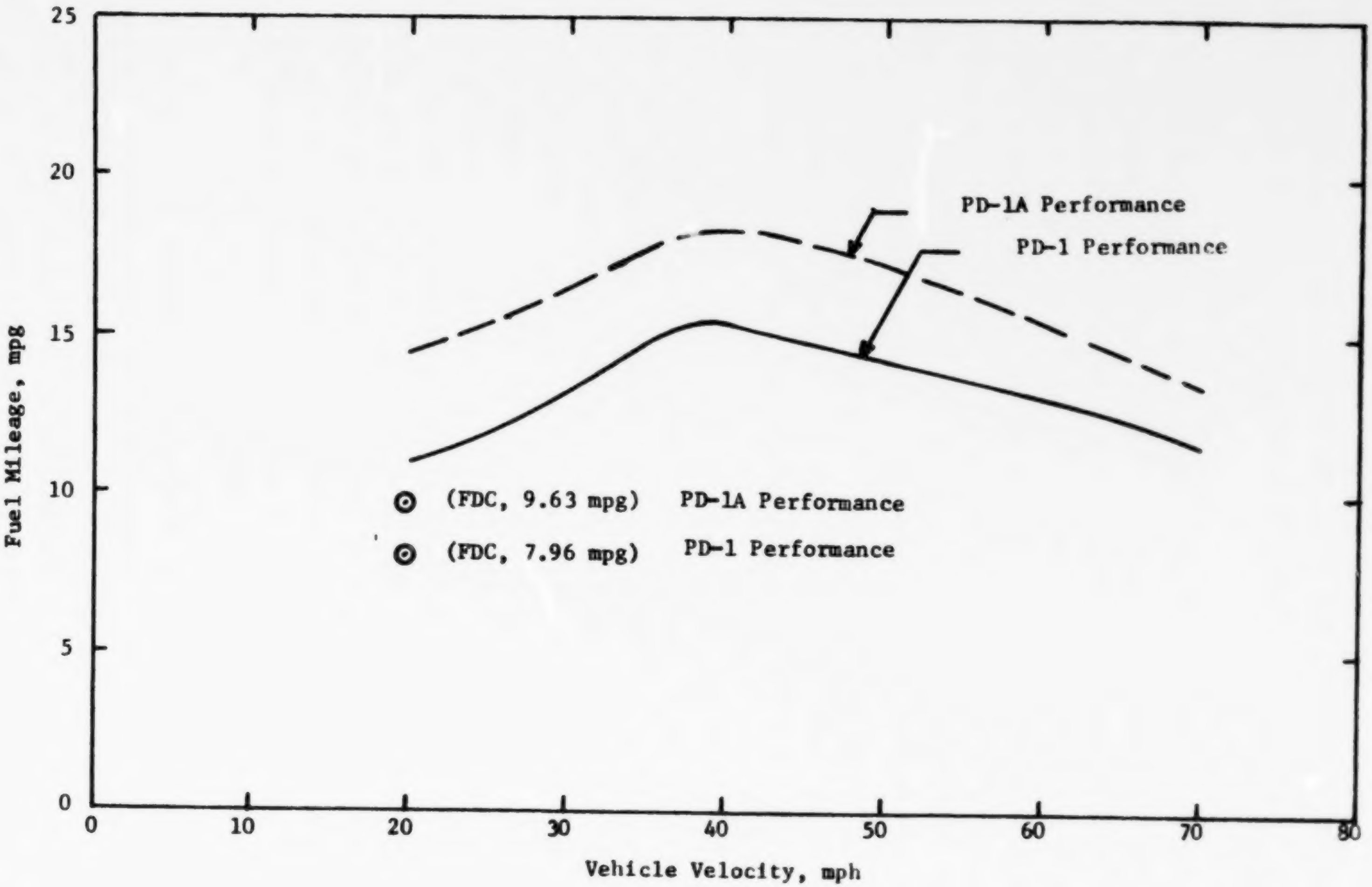


Figure 66. Comparison of PD-1 and PD-1A Engine Performance

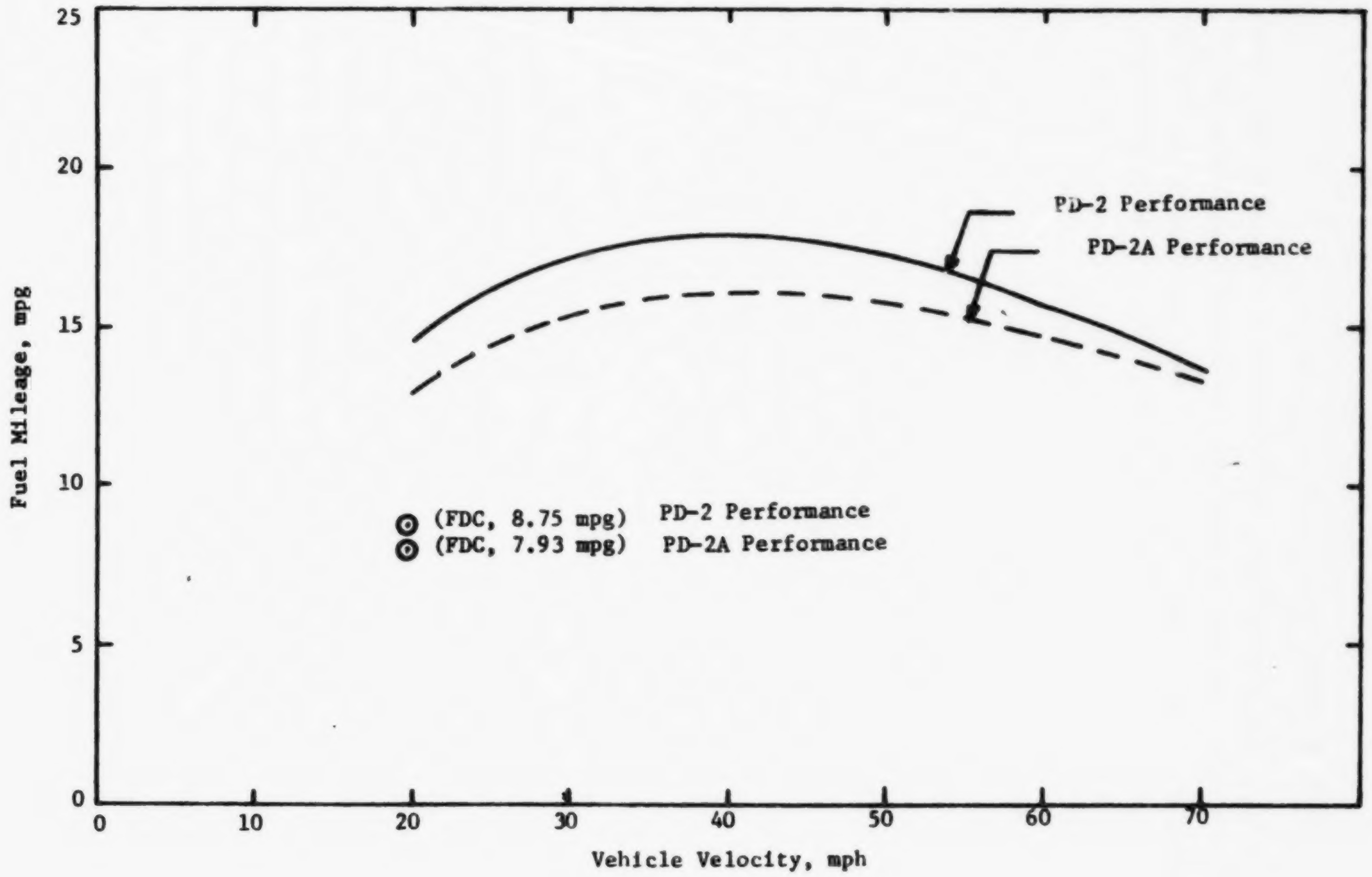


Figure 67. Comparison of PD-2 and PD-2A Engine Performance

3.6 Mission Analysis

Mission analysis includes the selection of the engine rated power level so that the vehicle can meet the performance specifications of reference 20 and the determination of the fuel consumed by the vehicle during the vehicle lifetime, 105,200 miles. The vehicle performance specifications are summarized in Table 16. Since the specifications for an ambient temperature of 105°F are more difficult to meet, the specifications for this temperature only were used. The Uniform Simplified Engine Duty Cycle shown in Table 17 was used to determine total fuel consumed. As in the case of the vehicle specifications, an ambient temperature of 105°F was used because fuel consumption increases with ambient temperature and this is the highest ambient temperature for which engine performance data were required. The Federal Driving Cycle (reference 21) referred to in Table 17 is a Department of Health, Education and Welfare urban dynamometer driving schedule for measuring automobile emissions; it gives vehicle velocity at one-second intervals for a total of 1370 seconds.

3.6.1 Analytical Methods

Computerized analytical methods were devised for engine transient analysis, engine wide open throttle acceleration and engine driving cycle analysis.

Engine Transient Analysis - The engine transient analysis code is a batch-type computer code with which the transient performance of the engine is simulated. Free power turbine and single shaft gas turbines were studied utilizing a digital dynamic system model. The dynamic models used in this project are full thermodynamic models of the appropriate cycle coupled with fuel control, a transmission, a drive train, and the vehicle dynamics, utilizing the SPADEI digital simulation language.

Each of the versions studied incorporated a rotating regenerator or recuperator for which a dynamic heat-lag representation was used (both hot and cold sides). Each version incorporated variable component geometry scheduled for optimum steady state performance (fuel consumption, turbine inlet temperature, and compressor stall margin), and opened

Table 16

Gas-Turbine Powered Vehicle Acceleration Specifications

Ambient Temperature, ^{o}F (Sea Level	Standing Start		Merging Traffic 25–70 mph, sec	DOT High-Speed Pass		30% Grade Velocity, mph	5% Grade Velocity, mph
	10 sec, ft	0–60 mph, sec		Time, sec	Maximum Distance, ft		
85^{o}F	440	13.5	15.0	15.0	1400	15	70
-20 to 105^{o}F (Excluding 85)	418	14.3	15.8	15.8	1470	14.2	69.5

Table 17

Uniform Simplified Engine Duty Cycle
(October 15, 1971)

	Average Speed, mph	Duration, hrs	Fraction of Life, %
Federal Driving Cycle	19.84	1750	50
Simplified Suburban Route	20	383	11
	30	384	11
	40	383	11
Simplified Country Route	50	200	5.67
	60	200	5.66
	70	200	5.67
Average/Totals	30	3500	100

quickly during accelerations. The fuel control schedule was optimized for best accelerations within stall margin and turbine inlet temperature constraints.

Each system was designed so that the maximum idle "creep" speed was less than 18 mph and the minimum top speed was greater than 85 mph. Two of the systems utilized a Tracor infinitely variable transmission. The gear ratio schedules were selected to give best vehicle acceleration characteristics taking power train inertia effects into consideration.

Horsepower scaling was handled by diameter scaling techniques for torque, flow, speed and inertia. The vehicle dynamics were as described in reference 20 with the following exceptions:

Inlet temperature, °F	105
Vehicle weight, lb	4600
Maximum tractive force, lb	2300
Accessory power, HP	4

The first system to be studied (CD-2) was a regenerative free-turbine engine with variable free-turbine inlet geometry. The free turbine was directly coupled (no fluid coupling, torque converter, or clutch) to the Tracor transmission and rear axle. For the system studied, an engine size of 150 horsepower (at 105°F) was necessary to meet the acceleration requirements.

The second system to be analyzed was a regenerated single-shaft engine (PD-1) with variable compressor outlet geometry and turbine inlet geometry. The power shaft was geared from the main shaft through the Tracor transmission, a two-speed transmission, a controlled-slipping clutch, and the rear axle. A rated power of 134 horsepower (at 105°F) was just sufficient to fulfill the acceleration requirements.

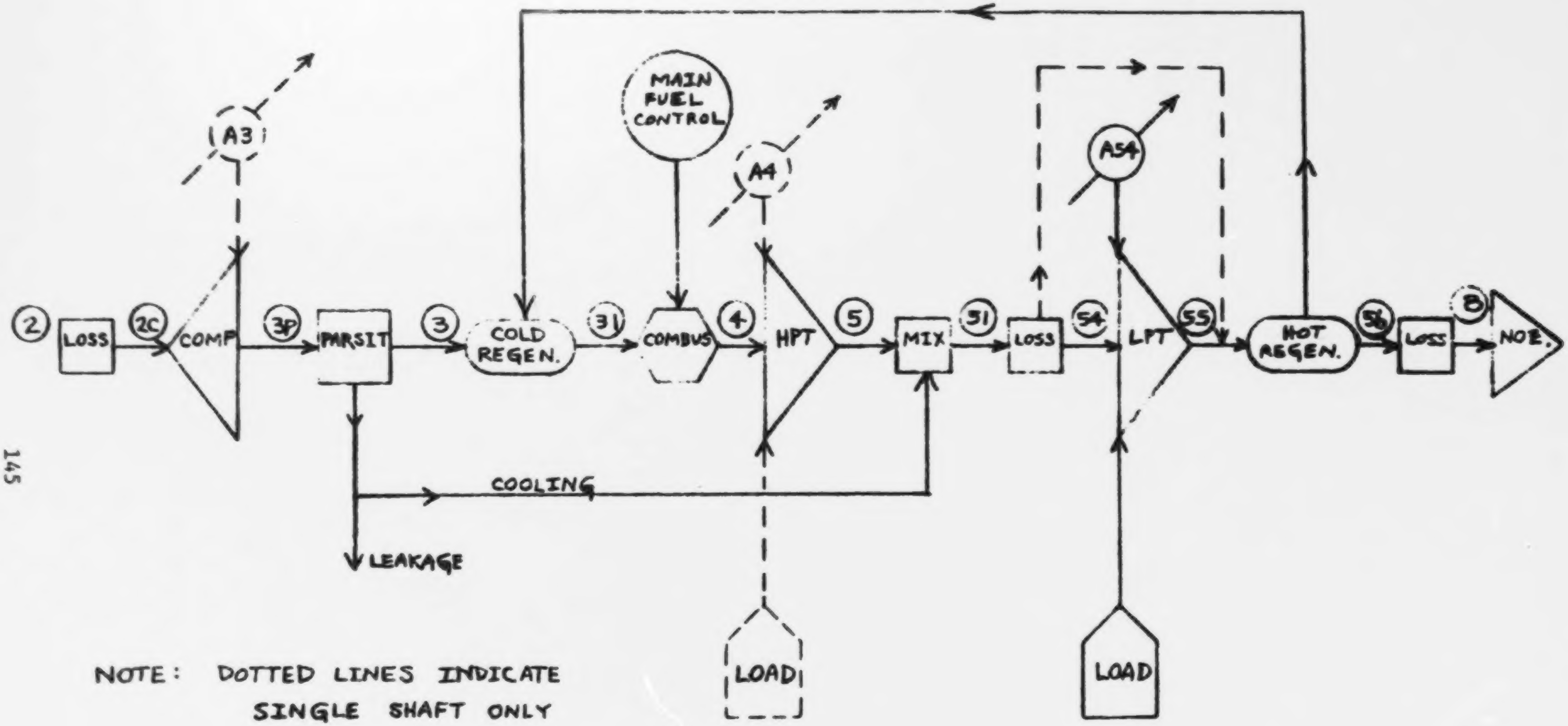
The third system (PD-2A)was a recuperated free-turbine engine with variable free-turbine inlet geometry. The free-turbine was connected to a torque converter, three-speed transmission, and rear axle. This system was studied by considering only the differences in the design-point kinetic energy of the two shafts compared to the CD-2 engine. The primary difference (compared to the first system) was a lighter power-turbine inertia. Thus, 134 horsepower (at 105°F) was sufficient for this advanced version.

The most difficult of the requirements for all systems turned out to be the acceleration distance in ten seconds. All other maneuvers (time to 60 mph, time from 25 to 70 mph, DOT passing maneuver, and grade climbing) were readily obtained.

The regenerated free-turbine version was the first system considered and will serve as the basis for discussion. The dynamic model of the engine is illustrated in Figure 68 with each engine component and station number designated for both the free power turbine and single shaft configurations. The model is similar to the steady state performance program and includes logic for the compressor, regenerator, combustor, turbines, and parasitic losses.

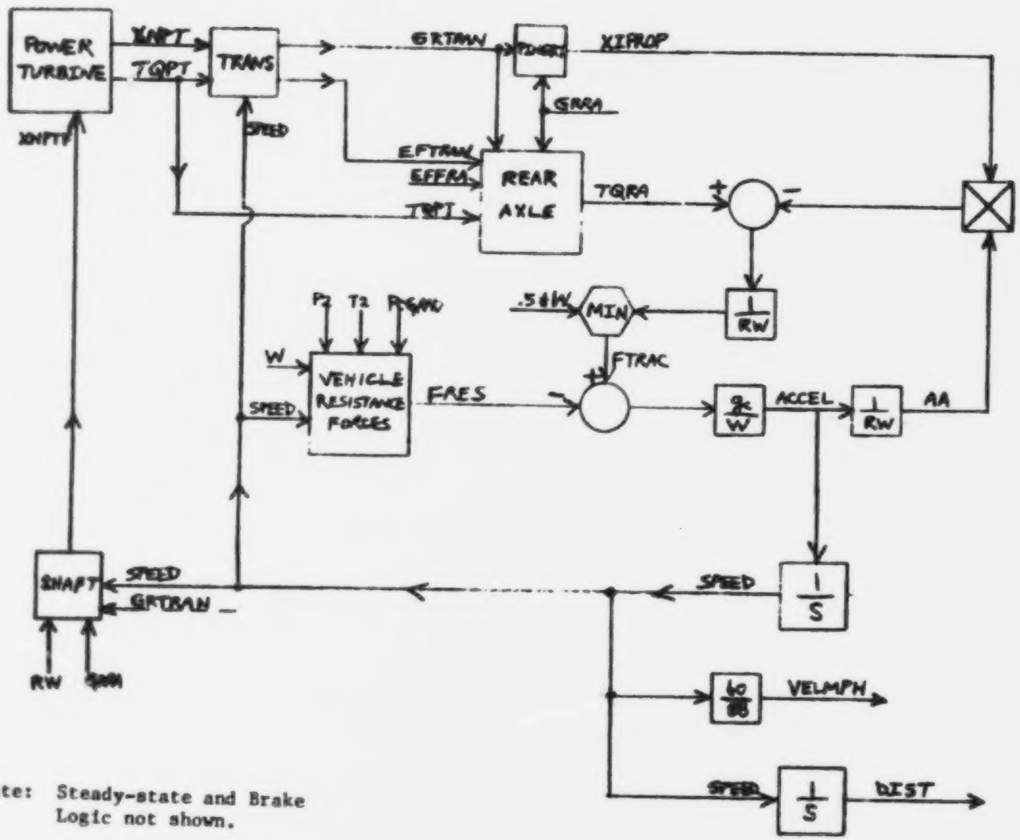
The power train and vehicle relationships are shown in Figure 69 for the free turbine, CD-2 engine. Power turbine speed and torque and vehicle speed are fed into the transmission logic with outputs of transmission gear ratio and efficiency. The gear ratio is used to calculate a drive train inertia, and the efficiency and power turbine torque are used to calculate torque at the rear axle, which is converted to tractive force. The vehicle resistance force is calculated from vehicle weight and velocity, and the difference between the tractive force and resistance force is used to calculate acceleration, including angular acceleration of the drive train. The vehicle acceleration is integrated to get speed, which is integrated to solve for distance travelled. The speed is fed back to the power turbine to permit an iterative solution. Similar relationships exist for the single shaft, PD-1, and free turbine PD-2 engines, with revisions for the clutch or torque converter.

The Tracor toroidal variable-ratio transmission efficiency (Figure 70) reflects vendor data down to about 8 horsepower. It was then necessary to make some horsepower loss assumptions in order to utilize the curve at zero power-turbine speed and horsepower. The same efficiency curve was also utilized for the single-shaft engines.


2
LOSS
2C
COMP
A3
3P
PARSIT
3
COLD REGEN.
31
MAIN FUEL CONTROL
COMBUS
4
A4
HPT
5
MIX
51
LOSS
54
A54
LPT
55
HOT REGEN.
56
LOSS
8
NOZ.
COOLING
LEAKAGE
LOAD
LOAD

NOTE: DOTTED LINES INDICATE SINGLE SHAFT ONLY

Figure 68. Automobile Gas Turbine - Dynamic Model Schematic


POWER TURBINE
XNPT
TQPT
TRANS
SPEED
GRTRAN
XIPROP
GRRA
EFTRAN
EFFRA
TQPT
REAR AXLE
TQRA
+
-
$\frac{1}{RW}$
.5 W
MIN
FTRAC
+
P2
T2
PCGRAD
W
SPEED
VEHICLE RESISTANCE FORCES
FRES
-
$\frac{g_c}{W}$
ACCEL
$\frac{1}{RW}$
AA
XNPT
SHAFT
SPEED
GRTRAN
RW
GRRA
SPEED
$\frac{1}{S}$
$\frac{60}{88}$
VELMPH
SPEED
$\frac{1}{S}$
DIST

Note: Steady-state and Brake Logic not shown.

Symbols

XNPT	power turbine speed
TQPT	power turbine torque
GRTRAN	transmission gear ratio
XIPROP	drive train inertia
GRRA	rear axle gear ratio
EFTRAN	transmission efficiency
EFFRA	rear axle efficiency
TQRA	rear axle torque
RW	wheel radius
FTRAC	tractive force
FRES	resistance force
W	vehicle weight
AA	angular acceleration
P2	compressor inlet pressure
T2	compressor inlet temperature
PCGRAD	percent grade

Figure 69. Free-Turbine Automobile Gas Turbine Power Train

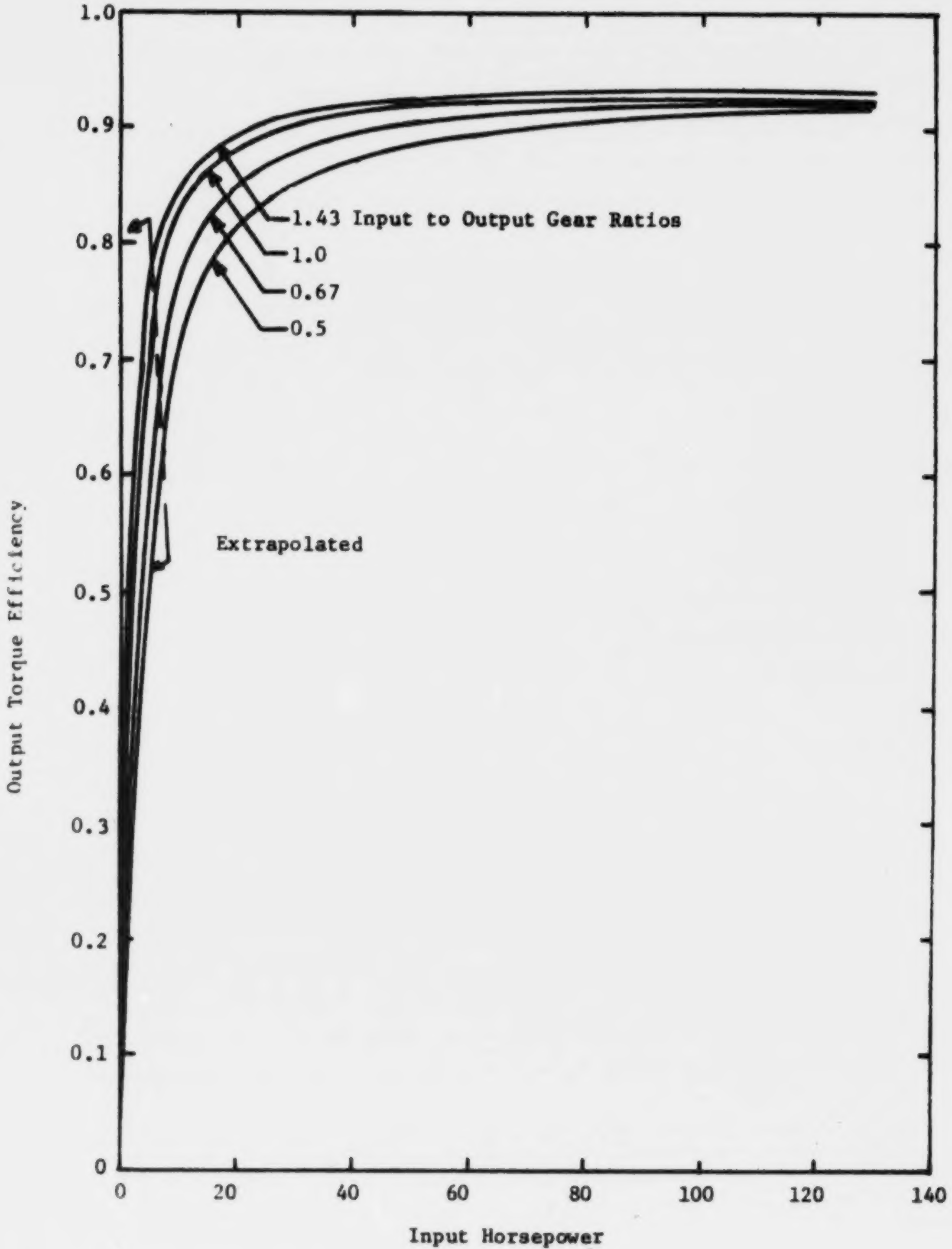


Figure 70. Tracor Infinitely Variable Transmission Efficiency

The variable power-turbine stator vanes are scheduled versus a corrected gas generator speed (Figure 71). This schedule is a very important factor in optimizing steady-state fuel consumption, turbine inlet temperature, and compressor stall margin. The large area scheduled at low speeds was needed for stall margin and idle maximum creep speed requirements.

Dynamically, it was found that the optimum vehicle performance was obtained by rapidly opening the vanes during accelerations. The dotted lines in Figure 72 illustrate the resultant stator vane control dynamics.

The fuel control (Figure 72) utilized for study was a typical flyball governor with maximum fuel schedule (fuel flow divided by compressor discharge static pressure) limitations.

When the flyball speed differs from the demanded gas generator speed, an error signal (epsilon one in Figure 72) is generated which, when integrated, causes a change in fuel valve position (X). This change is limited by a three-dimensional cam surface, called an acceleration schedule, which generates a maximum fuel flow (WFMAX), when multiplied logarithmically by the sensed compressor discharge pressure (PS3L). The XXDOT feedback path serves as a derivative compensation or anticipation element to minimize overshoot.

This same control, although a good choice for the free-turbine engine, was found to be very critical in the single-shaft engine and vehicle response to various loads, throttle demand rates, regenerator dynamics, and variable geometry dynamics. It is felt that a turbine inlet temperature control, such as those being studied for advanced military aircraft, would prove superior because of regenerator dynamics and load variations.

Vehicle decelerations utilizing engine braking effects and variable geometry were not studied, although the controls utilized did feature a minimum deceleration schedule to avoid a combustor flameout.

The dynamic characteristics of the CD-2 engine and vehicle are presented for acceleration on a level grade at 105 degrees Fahrenheit (Figures 73, 74 and 75). It should be noted that better performance would have resulted, if the reduced weight of the gas turbine and transmission (no

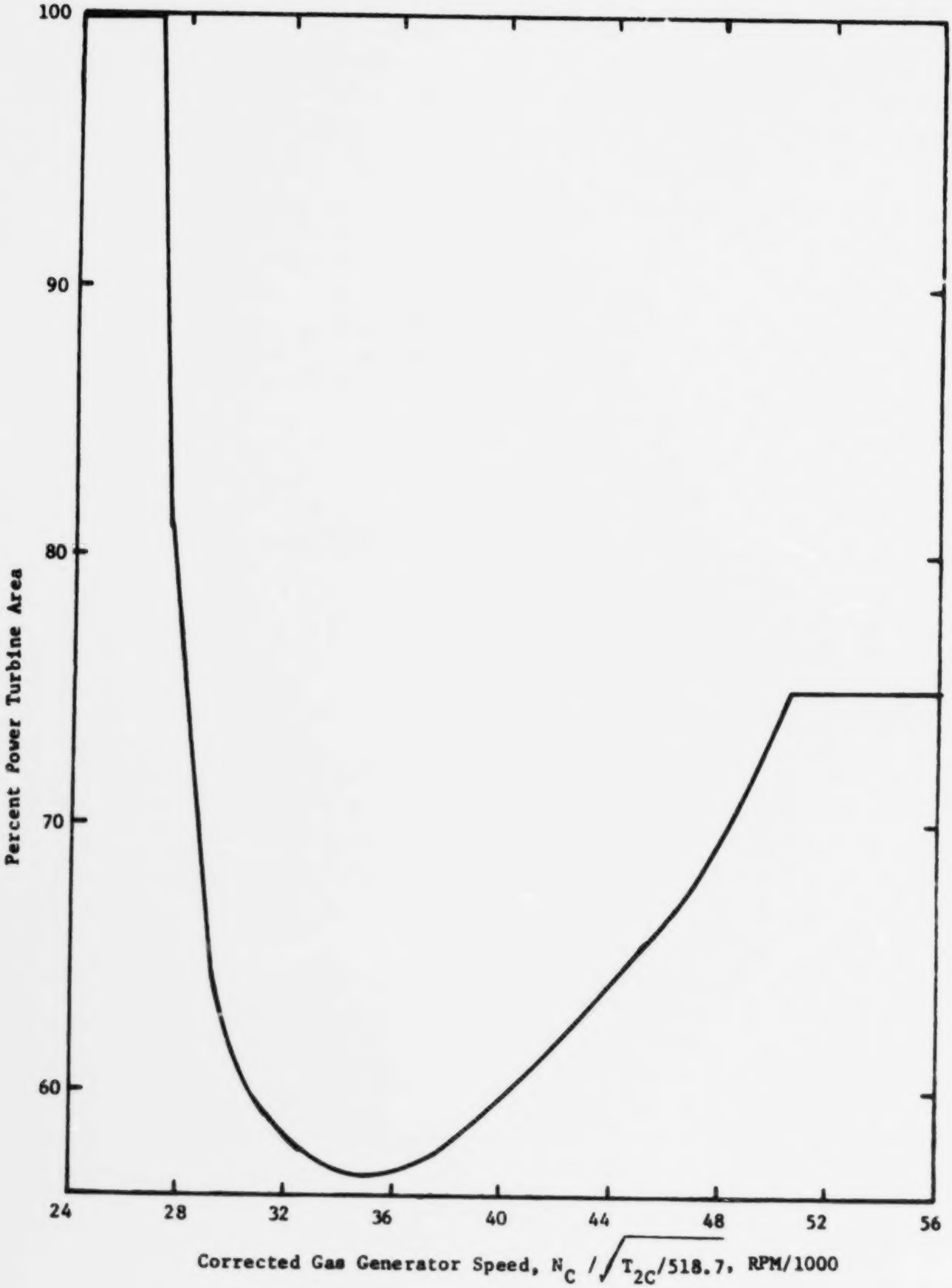
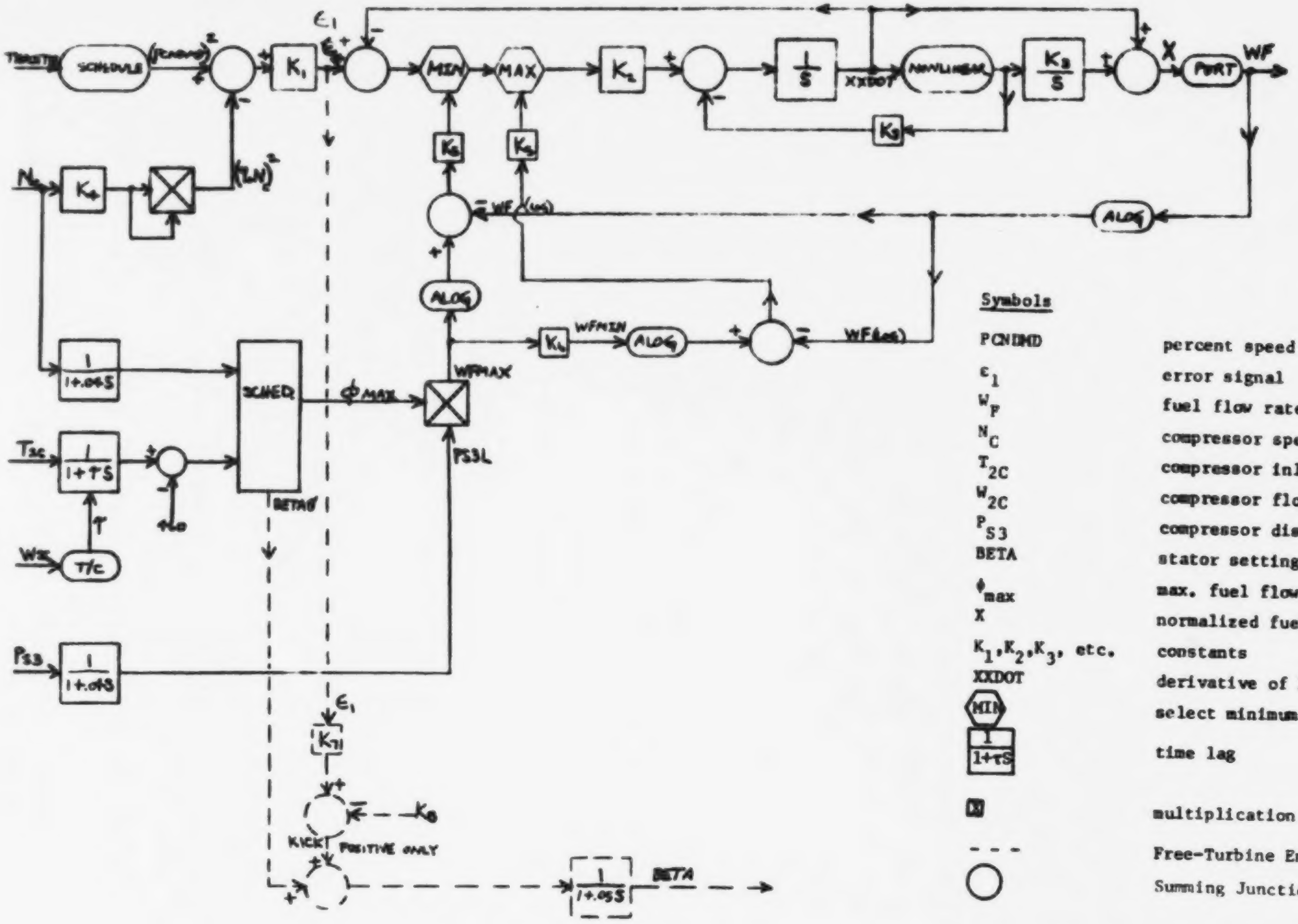


Figure 71. Free-Turbine CD-2 Engine Steady State Schedule Power Turbine Stators


THROTTLE
SCHEDULE
(PCNDMD)^2
+
-
K1
ε1
+
-
MIN
MAX
K2
1/S
XXDOT
NONLINEAR
K3/S
K3
X
PORT
WF
N
K4
(LN)^2
K6
K5
WF(ss)
ALOG
K6
WFMIN
WF(ss)
1/(1+.04S)
SCHED
φ MAX
WFMAX
T2C
1/(1+τS)
τ
W2C
T/C
BETA0
PS3L
PS3
1/(1+.04S)
ε1
K7I
K8
KICK
POSITIVE ONLY
1/(1+.05S)
BETA

Symbols

PCNDMD	percent speed demanded
ϵ_1	error signal
W_F	fuel flow rate
N_C	compressor speed
T_{2C}	compressor inlet temperature
W_{2C}	compressor flow rate
P_{S3}	compressor discharge pressure
BETA	stator setting
ϕ_{max}	max. fuel flow/static pressure, P_{S3}
X	normalized fuel valve position
K_1, K_2, K_3, etc.	constants
XXDOT	derivative of X
MIN	select minimum
$\frac{1}{1+\tau S}$	time lag
☒	multiplication $\boxed{1/S}$ integration
- - -	Free-Turbine Engine, Variable Power Turbine Stators
○	Summing Junction

Figure 72. Automobile Gas Turbine Fuel Control

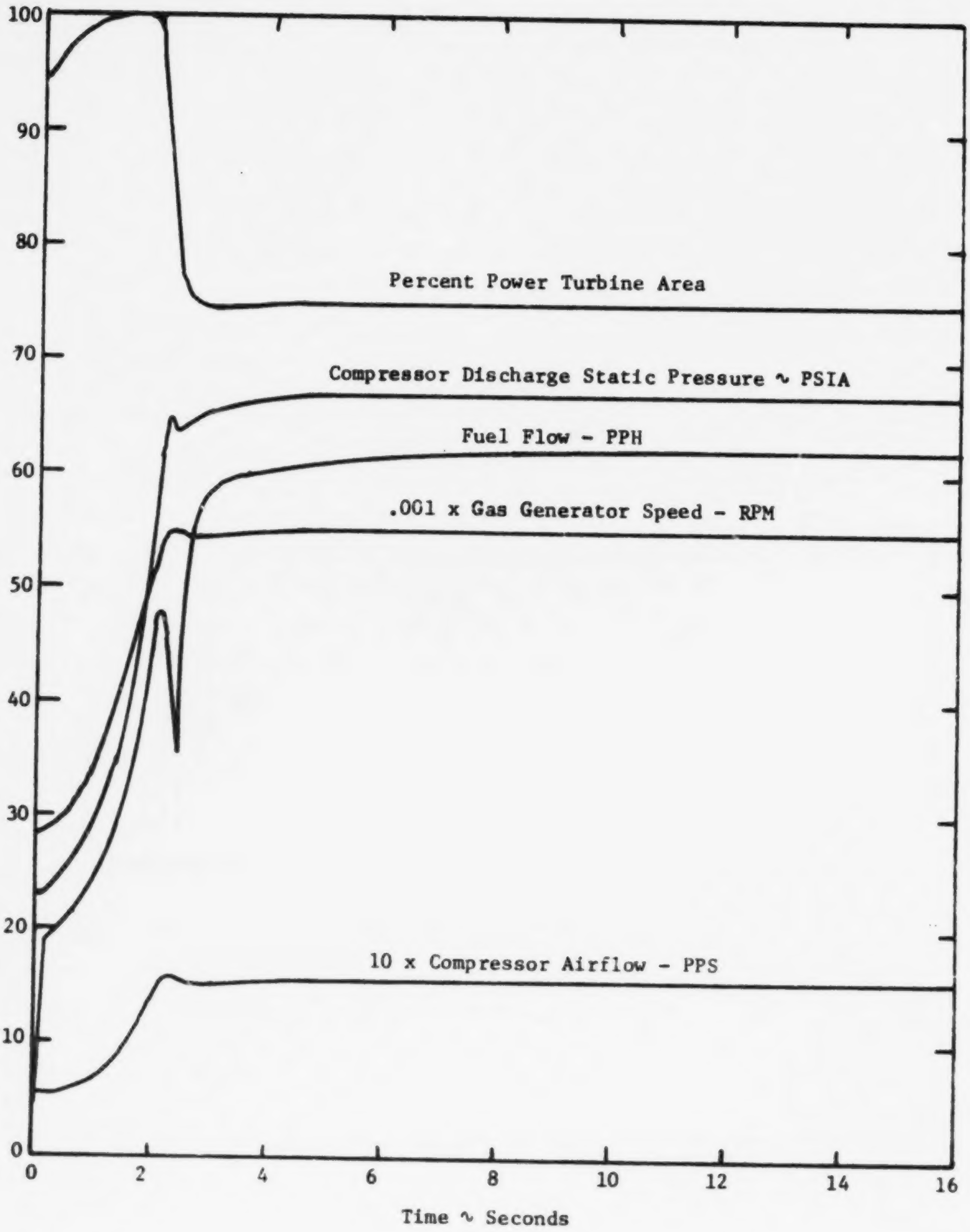


Figure 73. Free-Turbine CD-2 Engine Standing Acceleration - Level Grade

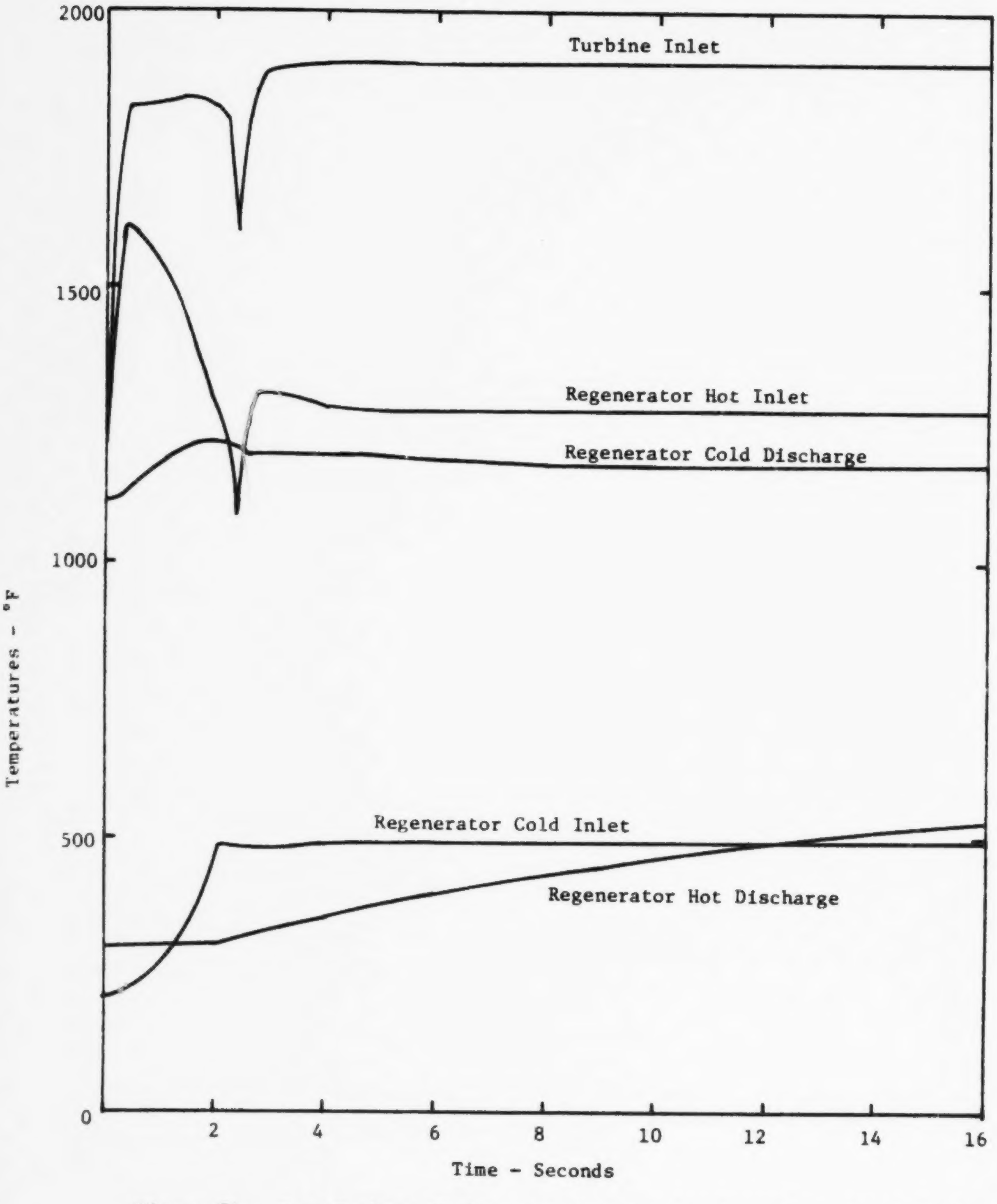


Figure 74. Free Turbine CD-2 Engine Standing Acceleration - Level Grade

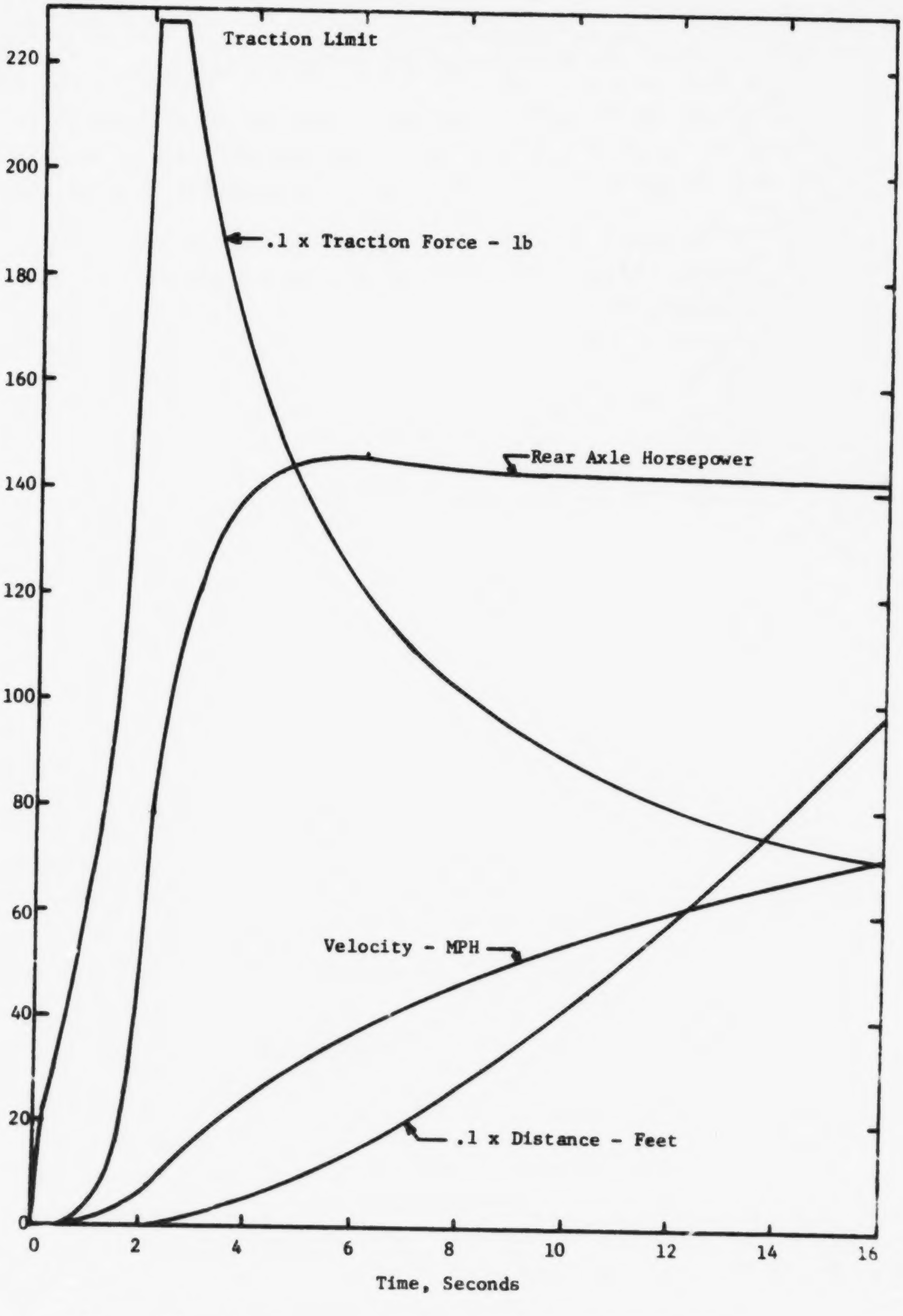


Figure 75. Free-Turbine CD-2 Engine Standing Acceleration, Level Grade

torque converter) had been deducted from the vehicle weight.

The DOT passing maneuver was assessed with the aid of Figure 76, showing the auto must travel 273 feet further than the truck (at 50 miles per hour) in less than 15.75 seconds, yet not exceeding 80 miles per hour.

The single-shaft engines (CD-1 and PD-1) incorporate a power train quite different in application. The variable-ratio transmission coupled with a two-speed transmission permits much more low-speed gearing without the power-train inertia becoming exorbitant, due to the slipping clutch. This gearing, plus the variable core geometry (See Figures 68 and 77), yields much quicker early response to a throttle demand - more traction force for the first two seconds.

However, the gearing and slipping clutch became a limiting item transiently when the clutch no longer slips, and thus limits the rear axle torque.

The resulting gear ratio dynamics necessary to maximize vehicle acceleration characteristics yield a very intricate control design, as depicted in Figure 78. A variable time constant is utilized, which is dependent on the vehicle velocity. These dynamics were optimized for level grade accelerations with a throttle actuation time of 0.5 second and the associated main fuel control optimized for engine dynamics (Figure 72). Other throttle actuation times, passenger loads, ambient conditions, or grades require other combinations of maximum fuel schedules and gear ratio dynamics in order to meet the requirements. The control system design, therefore, will be a very critical item in reaching a viable vehicle design. In contrast, the free-turbine engines, by nature, solve variations thermodynamically.

Figures 79 and 80 illustrate the standing acceleration response of the single-shaft PD-1 engines.

The free-turbine (PD-2A)engine was only briefly studied, for lack of time. Differences of rotor inertia and design speeds were considered and then applied as effective inertias to the first free-turbine model (CD-2). The primary difference was a reduced inertia for the power turbine, resulting in quicker engine and thus vehicle accelerations.

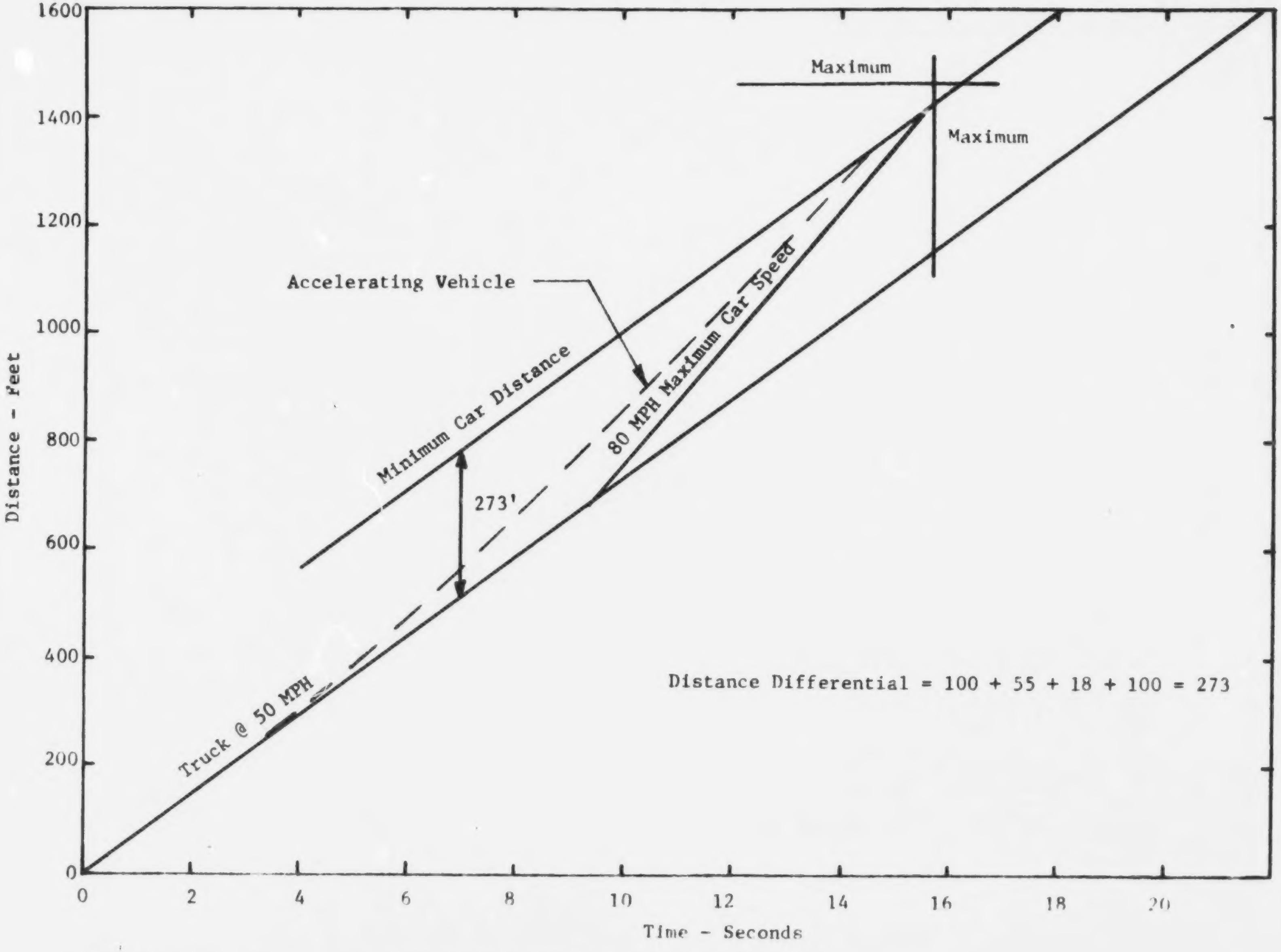


Figure 76. DOT Passing Maneuver Requirement, 105°F Day

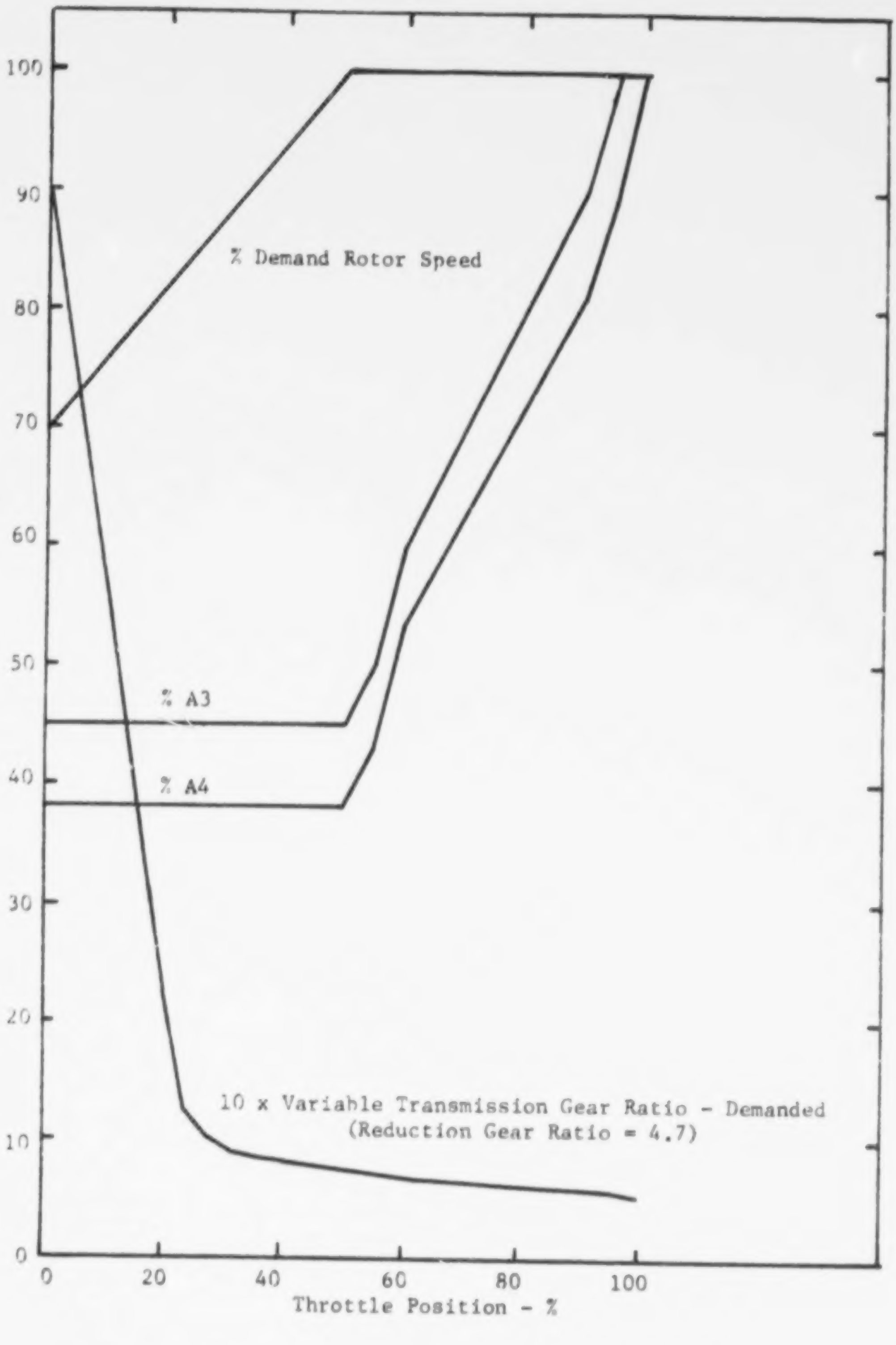
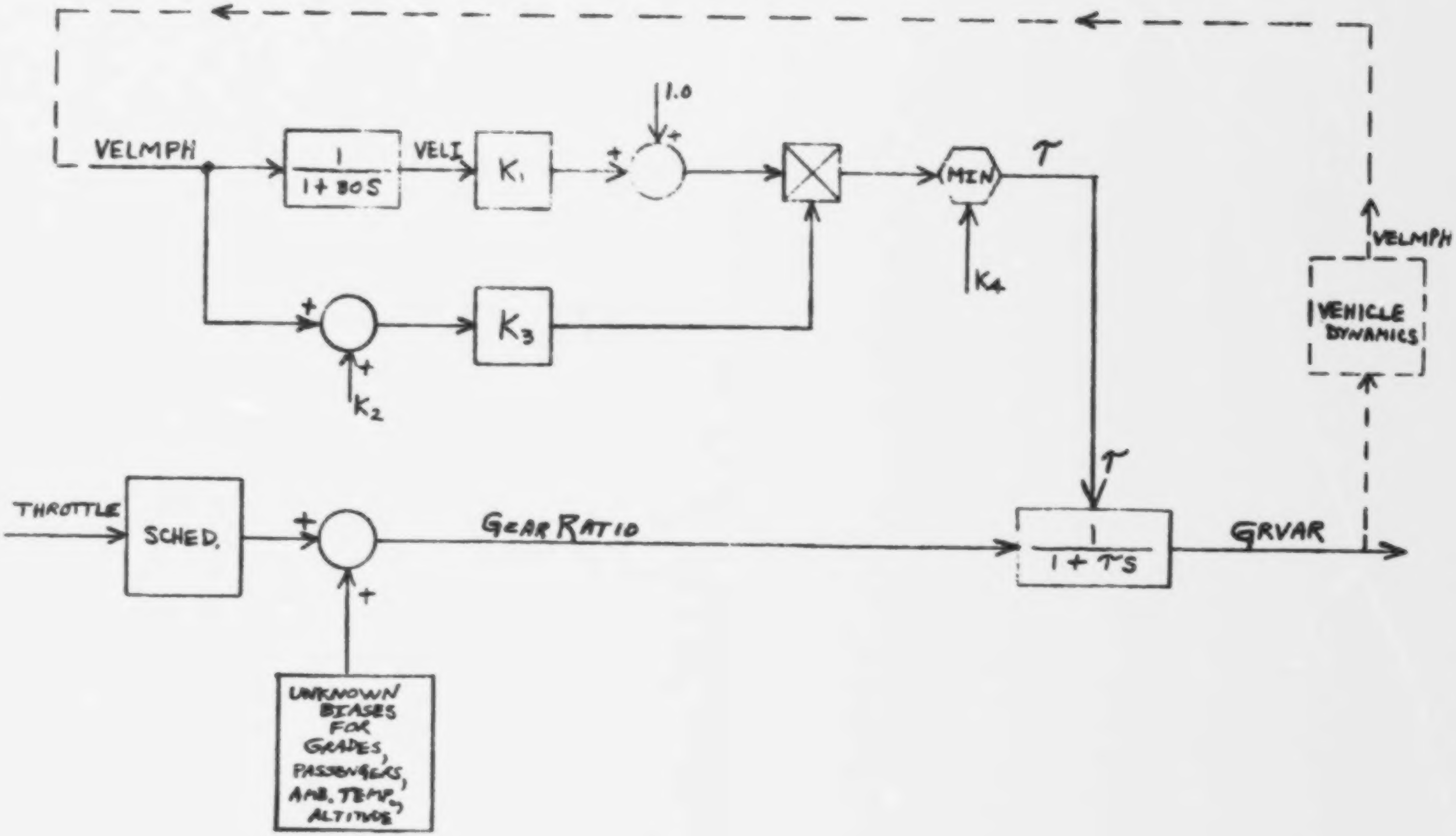


Figure 77. Single Shaft PD-1 Engine Throttle Schedules


VELMPH
1
1 + 80S
VELI
K1
1.0
+
+
MIN
T
K4
+
K3
+
K2
VELMPH
VEHICLE DYNAMICS
T
THROTTLE
SCHED.
+
GEAR RATIO
1
1 + TS
GRVAR
+
UNKNOWN BIASES FOR GRADES, PASSENGERS, AMB. TEMP., ALTITUDE

.igure 78. Single-Shaft PD-1 Engine - Preliminary Transmission Control

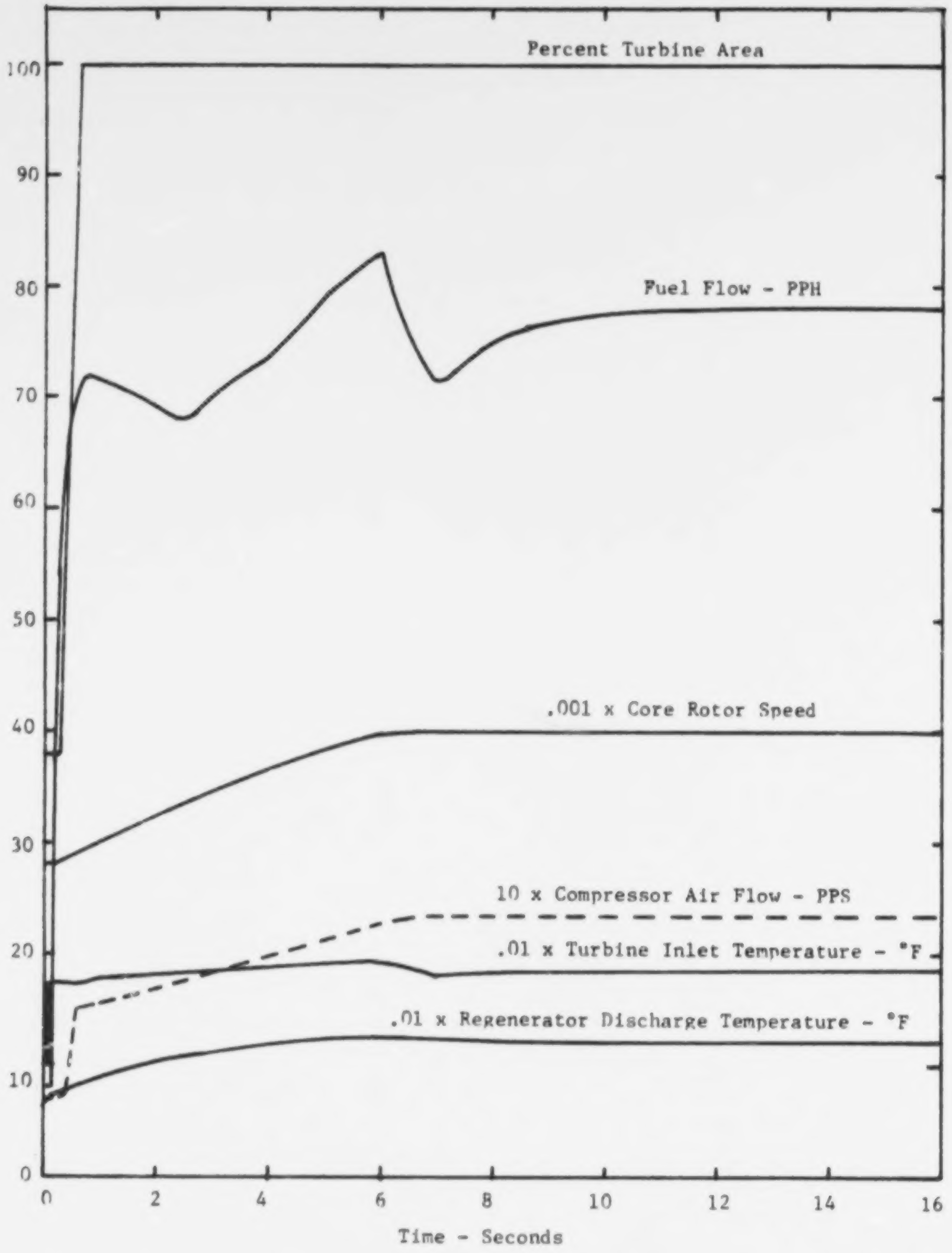


Figure 79. Single-Shaft PD-1 Engine Standing Acceleration - Level Grade

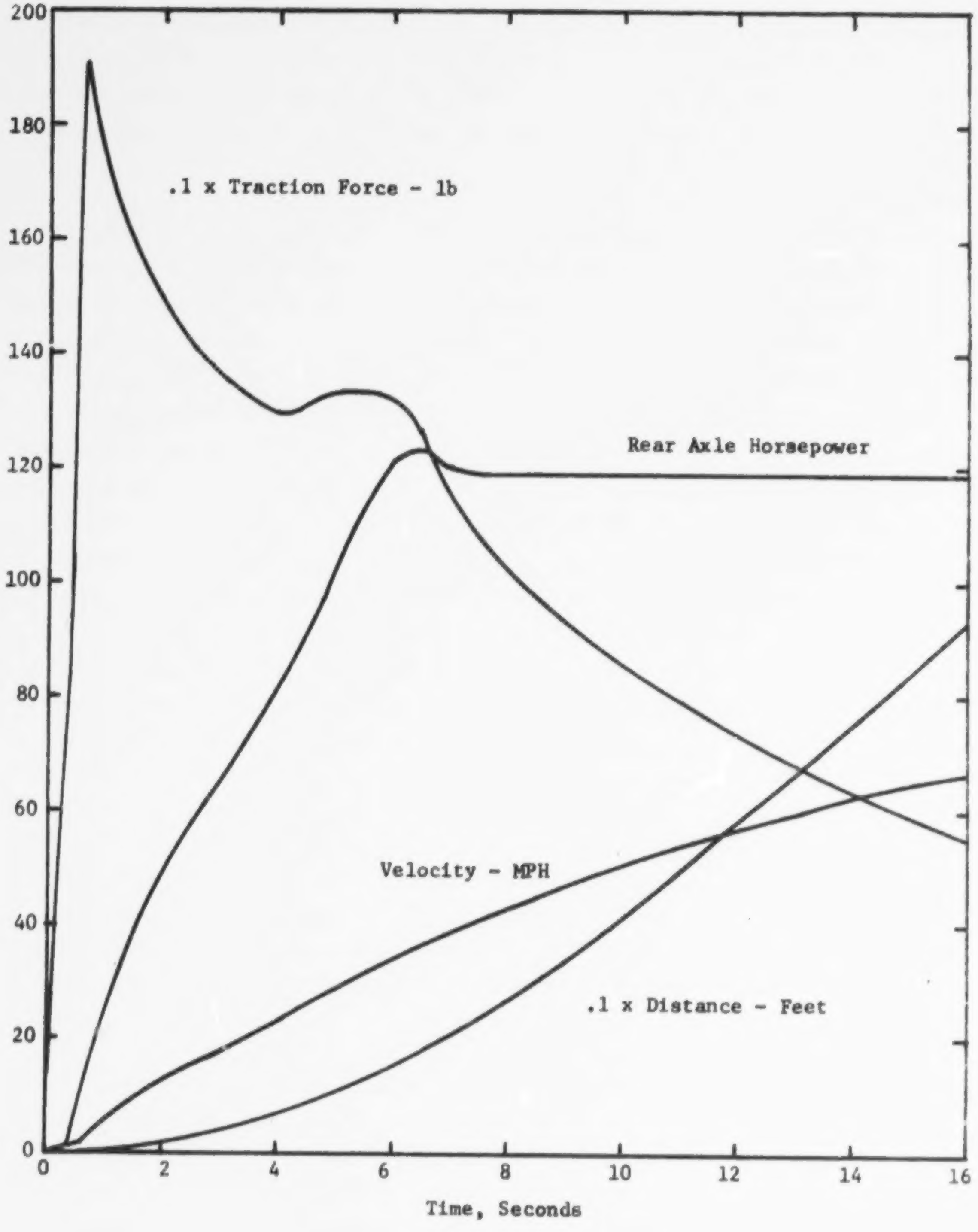


Figure 80. Single-Shaft PD-1 Engine Standing Acceleration Level Grade

The transient dynamic model was used to determine engine acceleration performance, however, the code is cumbersome to use because a detailed model of the control is required. The code is essential in studying different control philosophies, the first step in designing an engine control.

Vehicle Acceleration Analysis - For the determination of the vehicle performance during acceleration maneuvers listed in Table 16, a time sharing code which utilizes steady-state engine performance but accounts for the inertia of the engine rotor and drive train in determining the acceleration of the vehicle at wide-open-throttle (WOT) conditions was written. The control model used in this code is a mathematical specification of engine output speed with vehicle speed. The code integrates vehicle acceleration to get vehicle velocity and velocity to get distance, The program is flexible in that various engine speed-vehicle velocity schedules, transmission characteristics and acceleration modes can easily be inserted. Provisions are made for scaling engine torque and speed proportional to selected values at rated conditions. Although the integration interval is smaller the code prints out vehicle and engine speeds, vehicle acceleration and distance traveled at one-second intervals. This code was especially useful in analyzing single-shaft engines in which there is no gas generator.

Engine Driving Cycle Analysis - The determination of fuel consumption at the six steady state speeds in the Uniform Simplified Engine Duty Cycle, Table 17, is made directly from the engine performance maps taking into account the applicable rolling and aerodynamic resistance of the vehicle and the speed ratios, shift points and efficiency values for the transmission and final drive.

In the case of the Federal Driving Cycle, Table 17, a Mission Analysis computer code determines the required vehicle acceleration for each of the one-second intervals of the Federal Driving Cycle (FDC). Figure 81 is a histogram which shows the envelope of maximum tractive effort required for a 4600-pound vehicle on the FDC as a function of vehicle velocity. The tractive effort is the sum of the acceleration force and resistance forces for the vehicle. The velocity is given and the acceleration is implicit in the FDC data. The code determines the total fuel consumed

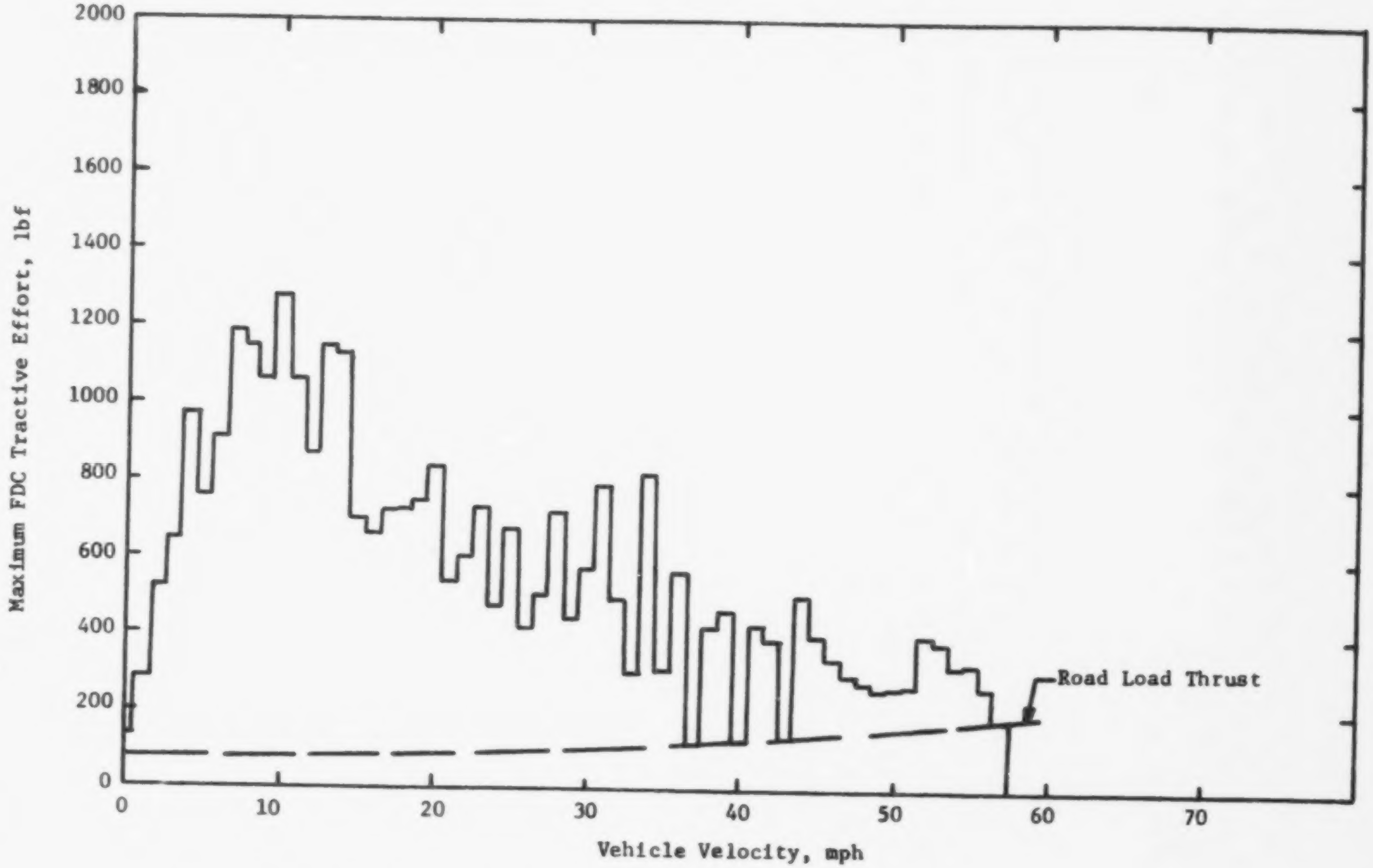


Figure 81. Envelope of Maximum Tractive Effort on FDC for a 4600-pound Vehicle

on the FDC from these data and the input characteristics of the engines.

The features of the Mission Analysis Code include:

- Second-by-second determination of vehicle acceleration on the FDC.
- Marching-time solution with an account of engine and drive-train inertia.
- Engine operation near the specified operating line. (Operation off this line occurs wherever torque to accelerate the engine is needed)
- When the engine is decelerating but the vehicle requires a positive tractive effort the engine rotative energy difference produces tractive power.
- Full torque is assumed to be available over the whole interval because the stator vanes and control valves are assumed to have negligible inertia compared to the engine rotor.

For each one-second interval the code calculates and/or stores the following quantities:

Time
Vehicular velocity
Distance Traveled
Acceleration for the interval
Propulsion power required (negative values indicate braking)
Engine output shaft speed
Engine output torque
Fuel flow rate
Gear ratio
Tractive effort
Road-load thrust
Transmission input torque
Transmission output torque
Axle torque

A separate program accesses the stored data and prints out evaluative data, such as, the total time the engine was required to produce preset levels of power, etc. Printed out at the end of each computer run are:

Total fuel used
Fuel mileage
Average engine horsepower
Average engine speed
Average vehicle velocity
Maximum engine power
Maximum engine speed
Maximum engine torque

3.6.2 Mission Analysis Engines

The salient features of the seven engines examined by the Mission Analysis code are given in Table 18. The group of 7 engines consist of 4 General Electric engines (CD-1, CD-2, PD-1A, PD-2A), one from each of 2 EPA contractors, plus an internal combustion engine. Design data for CD-1 and CD-2 may be found in Table 14 and for PD-1A and PD-2A in Table 15. The following is a brief discussion of the drive-trains of each of the first five engines under consideration; the two engines from the EPA contractors are described in Appendix B which will be published separately.

CD-1 Single Shaft Engine - Figure 82 is a schematic of the drive train selected for the CD-1 (Single Shaft) engine. A "Tracor" transmission whose characteristics were previously given in Figure 73 and are tabulated in Table 19 is used. Also given in Table 19 are the limiting values of the B_{10} roller life in hours; the B_{10} life is a statistical life below which no more than 10% of the units will have failed. A slipping clutch allows for continuity between the non-zero engine idle speed of 70% of design speed and zero vehicle velocity. Although the slipping clutch allows for a speed discontinuity between the engine shaft and the axle, full torque is transmitted across this clutch. In addition, a two-speed transmission is included to minimize the vehicle velocity range in which slipping occurs. With the gearing, slip can occur up to a vehicle velocity of 8.15 mph. However, because of the selected engine speed-vehicle velocity paradigms, slip occurs only up to a vehicle velocity

Table 18

Mission Analysis Engines

Description	Pressure Ratio	Moment of Inertia, lb ft sec^2	Reduction Gear Ratios	Rear Axle Ratio	Max. Engine Speed, RPM	Coupling	Transmission	Transmission Gear Ratios	Auxiliaries	Source
CD-1 Single Shaft Regenerative Variable Compressor and Turbine	3.6:1	0.012	4:1 and 3.69:1	2.9:1	40000	Slipping Clutch	Infinitely Variable	3:1 to 1:1.5	Low (3:1) range High (1:1) range	Preliminary Selection
CD-2 Free-Turbine Regenerative Variable Power Turbine	5:1	0.00249 0.01	3.5:1 and 3.69:1	2.9:1	55000 G.G. 35000 P.T.	None	Infinitely Variable	3:1 to 1:1.5	None	Preliminary Selection
PD-1A Single Shaft Regenerative Bypass Combustor Variable Compressor and Turbine	3.2:1	0.012	4:1 and 3.69:1	2.9:1	40000	Slipping Clutch	Infinitely Variable	3:1 to 1:1.5	Low (3:1) range High (1:1) range	Preliminary Design
PD-2A Free Turbine Recuperated 1000°F Combustor Variable Power Turbine	6.6:1	0.001 0.00124	12.5:1	2.57:1	80000 (GG) 50000 (PT)	Torque Converter	Automatic	2:45/1.45/ 1.0:1	None	Preliminary Design
Single Shaft Simple cycle	10:1	0.00065	37:1	2:1	100000	None	Hydromechanical Infinitely Variable	∞ to 0.8:1	None	United Aircraft
Single Shaft Regenerative Variable IGV		0.0015	7.91:1	16:1	79100 83050 (WOT)	Slipping Clutch	Infinitely Variable	3:1 to 1:2.0	None	AiResearch
Internal Combustion Engine		0.45	None	2.91:1	4000	Torque Converter	Automatic	2.45/1.45/ 1:1	None	EPA

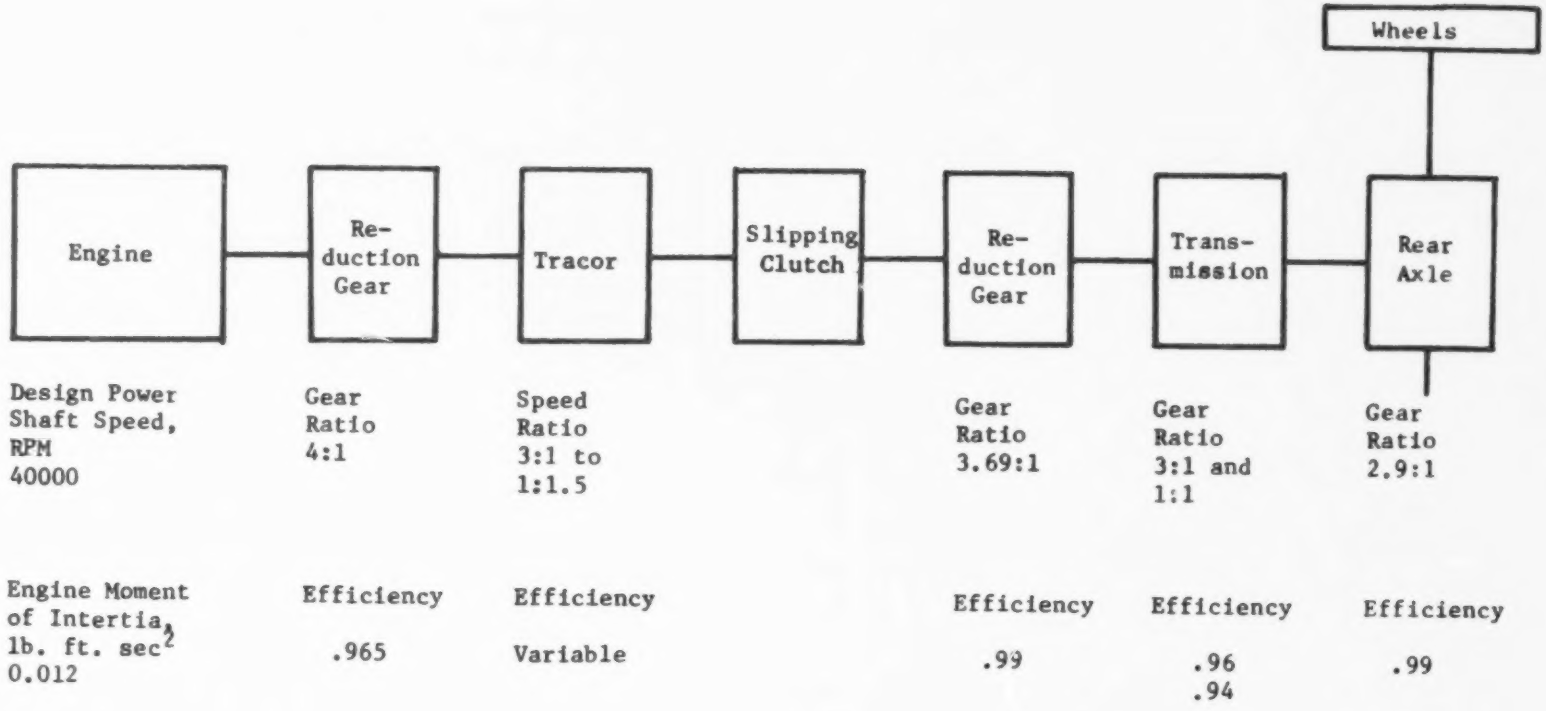

Engine
Re-duction Gear
Tracor
Slipping Clutch
Re-duction Gear
Trans-mission
Rear Axle
Wheels
Design Power Shaft Speed, RPM 40000
Gear Ratio 4:1
Speed Ratio 3:1 to 1:1.5
Gear Ratio 3.69:1
Gear Ratio 3:1 and 1:1
Gear Ratio 2.9:1
Engine Moment of Intertia, lb. ft. sec^2 0.012
Efficiency .965
Efficiency Variable
Efficiency .99
Efficiency .96 .94
Efficiency .99

Figure 82. Schematic of Drive Train for CD-1 (Single-Shaft) Engine

Table 19

Calculated "Tracor" Transmission Data Points (150 HP)

(5:00" Diameter Toroid, Two 3.4" Diameter Power Rollers)

Drive Ratio	3:1	2:1	1.52:1	1.32:1	1:1	1:1.5
150 HP at 10,000 RPM						
Life (B_{10}, hours)	4	49	139	211	320	231
Efficiency (percent)	93.7	94.1	93.7	93.2	92.2	89.4
49.2 HP at 8,900 RPM						
Life (B_{10}, hours)	193	2,749	8,360	12,799	19,863	14,744
Efficiency (percent)	90.6	90.6	89.9	89.1	86.9	81.8
34.8 HP at 8,300 RPM						
Life (B_{10}, hours)	554	8,435	25,800	39,124	57,274	40,027
Efficiency (percent)	89.1	89.2	88.3	87.4	84.9	78.7
24 HP at 7,500 RPM						
Life (B_{10}, hours)	1,578	25,268	66,692	91,712	124,758	86,691
Efficiency (percent)	87.3	87.4	86.5	85.4	82.6	75.6
16.1 HP at 6,700 RPM						
Life (B_{10}, hours)	4,977	64,014	148,085	201,726	276,289	194,090
Efficiency (percent)	84.8	85.0	83.9	82.8	79.5	71.7

Table 19 (Cont'd.)

Calculated "Tracor" Transmission Data Points (150 HP)

(5:00" Diameter Toroid, Two 3.4" Diameter Power Rollers)

Drive Ratio	3:1	2:1	1.52:1	1.32:1	1:1	1:1.5
10.1 HP at 6,100 RPM						
Efficiency (percent)	80.4	80.6	79.2	77.7	73.6	63.5
6.3 HP at 5,800 RPM						
Efficiency (percent)	73.3	73.2	71.1	69.1	63.5	49.5

of 6.52 mph during wide-open-throttle maneuvers and during operation on the FDC.

Shown in Figure 83 is the engine speed-vehicle velocity paradigm for the CD-1 engine used for the FDC. The engine speed, N_E, is normalized with the design speed, $N_{E,D}$, the vehicle velocity, V_V, is normalized with maximum vehicle velocity, V_{VM}, obtainable with the transmission in the highest speed ratio and the tractive effort, F_T, is normalized by the maximum required tractive effort, $F_{T,M}$, on the FDC. The figure is a sketch of the following linear equation.

$$\frac{N_E}{N_{E,D}} = \frac{N_{ID}}{N_{E,D}} + B^* \left[\frac{F_T}{F_{T,M}} \ \frac{V_V}{V_{VM}}\right] \qquad (19)$$

where N_{ID} is the engine idle speed and B^* is a constant which is adjusted once for each engine so that the engine will just negotiate all of the accelerations called for on the FDC at the lowest engine speed. As the vehicle velocity, V_V, and the tractive power, $F_T V_V$, increases the engine speed, N_E, increases to provide the required power. All operation to the left of the lock up locus is accomplished with slip in the clutch.

Shown in Figure 84 is the engine speed-vehicle velocity paradigm for wide-open-throttle (WOT) acceleration, a sketch of the following equation.

$$Y = A + BX + CX^2 + DX^3 \qquad (20)$$

where

$$Y = \frac{N_E}{N_{E,D}} \quad , \quad X = \frac{V_V}{V_{VM}} \qquad \text{and}$$

A, B, C and D are assigned constants. These constants are chosen to give the vehicle smooth acceleration and to minimize the region over which slip occurs. Referring to the paradigm in the figure, slip occurs in Region I. When the vehicle velocity is in the range of Regions II and III the slipping clutch is locked and the infinitely variable transmission begins reducing speed ratio from the highest ratio of 3:1 toward 1:1. When 1:1 is reached the infinitely variable transmission returns

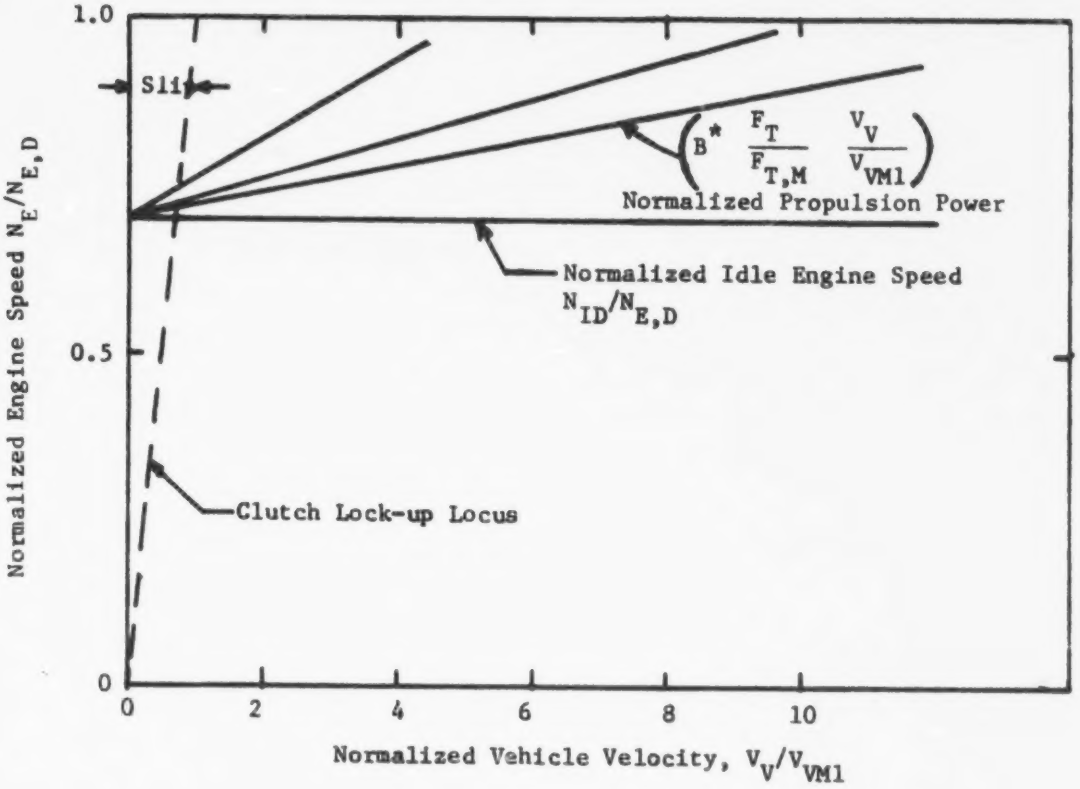


Figure 83. Schematic of Federal Driving Cycle Engine Speed-Vehicle Velocity Paradigm for Single Shaft Engine

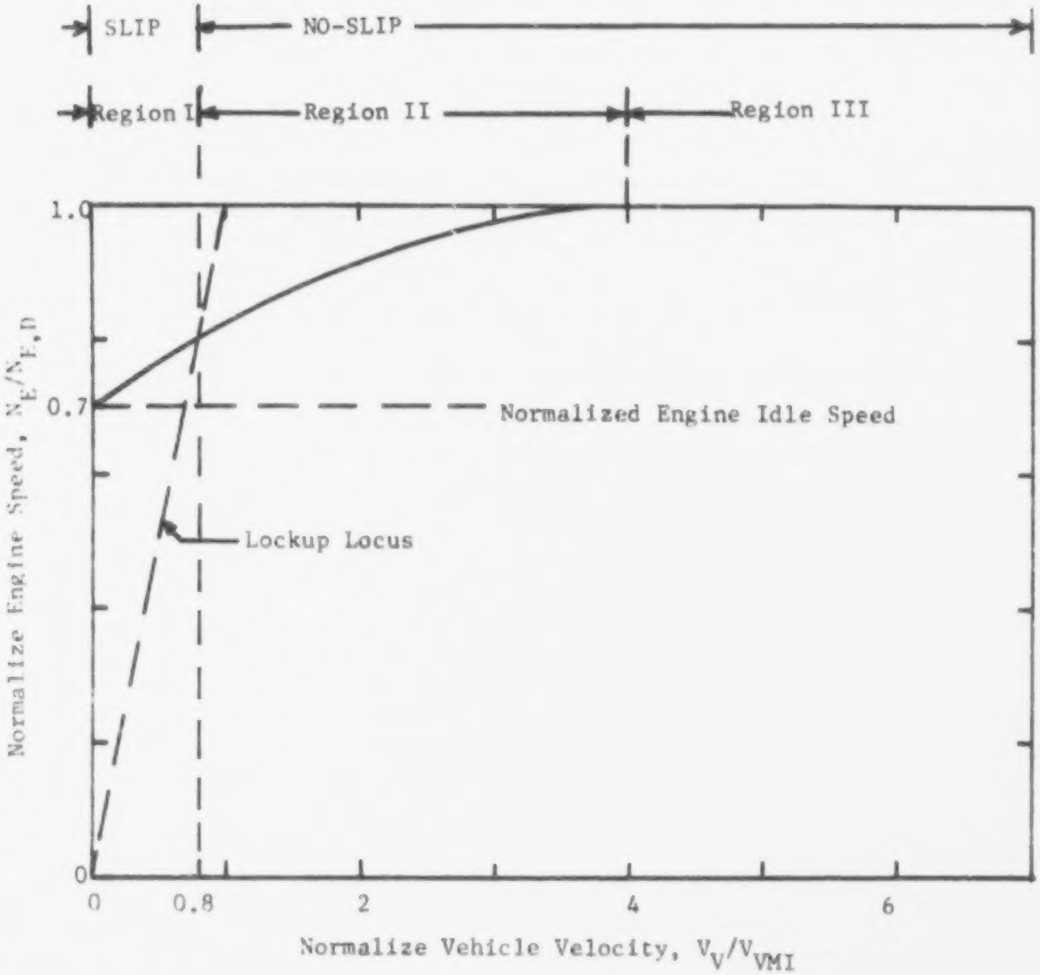


Figure 84. Schematic of Wide Open Throttle Engine-Speed-Vehicle Velocity Paradigm for Single-Shaft Engine

to a speed ratio of 3:1 simultaneously with the two-speed transmission shifting from 3:1 to 1:1. The engine continues to accelerate until it reaches design speed. At this point the infinitely variable transmission continues to reduce speed ratio until at maximum vehicle speed the ratio is 1:1.5. The constants used for the CD-1 engine in Equation 20 for the three regions are:

	Region I $(0 \leq X \leq 0.8)$	Region II $(0.8 \leq X\ 4.0)$	Region III $(X \geq 4.0)$
A	0.70	0.6875	1.0
B	0.125	0.15625	0.0
C	0.0	-0.01953	0.0
D	0.0	-7.567×10^{-10}	0.0

This selection of constants results in a match of coordinates and slopes at the boundaries of the regions. Furthermore, slip occurs in the clutch for $X < 0.8$, or up to 6.52 mph since V_{VM1} equals 8.15 mph for the gearing on the CD-1 engine.

In addition to the 0-60 mph WOT acceleration, runs were made to simulate the DOT passing maneuver and the merging maneuver. At 50 mph on the DOT maneuver, the steady-state road load thrust requires about 25 horsepower, depending upon the particular transmission. For the single-shaft engines (CD-1 and PD-1) which both have 70% idle speeds, this 25 horsepower can be obtained by running the engine at 70% engine speed with the vanes opened. Furthermore, this condition of 25 HP at 70% engine speed is on the line of minimum fuel consumption, and is taken as the initial condition for the DOT passing maneuver. Then, when the maneuver begins, both the engine speed and vehicle velocity must be increased. Several different schedules of engine speed versus vehicle velocity were employed to examine their effect on distance traveled during the maneuver. A parabolic relationship of this form was used:

$$\frac{N_{PT}}{N_{DPT}} = a + b\ V_V + c\ V_V^{\ 2}$$

The constants were evaluated using these boundary conditions:

I. 70% engine speed at 50 mph cruise velocity

$$\frac{N_{PT}}{N_{DPT}} = 0.7 \text{ at } V_V = 50 \text{ mph}$$

II. 100% engine speed and zero slope at velocity V_*

$$\left.\begin{aligned} \frac{N_{PT}}{N_{DPT}} &= 1.0 \\ \frac{d(N_{PT}/N_{DPT})}{d\,V_V} &= 0.0 \end{aligned}\right\} \text{at } V_V = V_*$$

Velocity V_* is that velocity where the engine speed reaches 100% for the DOT passing maneuver. The parameter V_* was set at 53 mph.

A similar approach was used for the merging maneuver, where the cruise velocity is 25 mph and V_* was set at 28 mph.

PD-1A Single Shaft Engine - Figure 85 is a schematic of the drive train selected for the PD-1A (Single Shaft) engine. The same "Tracor" transmission as was used on engine CD-1 is used for engine PD-1A; the engine idle is 70% of design speed. A slipping clutch is also used, and the discussion of the slipping clutch and the engine speed-vehicle velocity paradigms given for engine CD-1 are applicable to engine PD-1A. Both the engine transient analysis described earlier in Section 3.6.1 and the analysis described above for engine CD-1 were used to calculate the WOT acceleration performance of engine PD-1A.

CD-2 Free Turbine Engine - Figure 86 is a schematic of the drive train selected for the CD-2 (Free Turbine) engine. A "Tracor" infinitely variable transmission (IVT) couples the engine to the wheels. Table 19 lists the calculated values of efficiency of a Tracor transmission at a 150 HP rating (reference 18); this efficiency data was shown in Figure 13. Two reduction gears are required because the rated input speed of the IVT is too high (10,000 RPM). For the Federal Driving Cycle, the engine speed was set by the following procedure. The horsepower corresponding to the road load thrust was determined. Then, the engine speed was set at that horsepower on the line of minimum specific fuel consumption. The total horsepower was then calculated as the sum of that for road load thrust and of that for engine rotative acceleration.

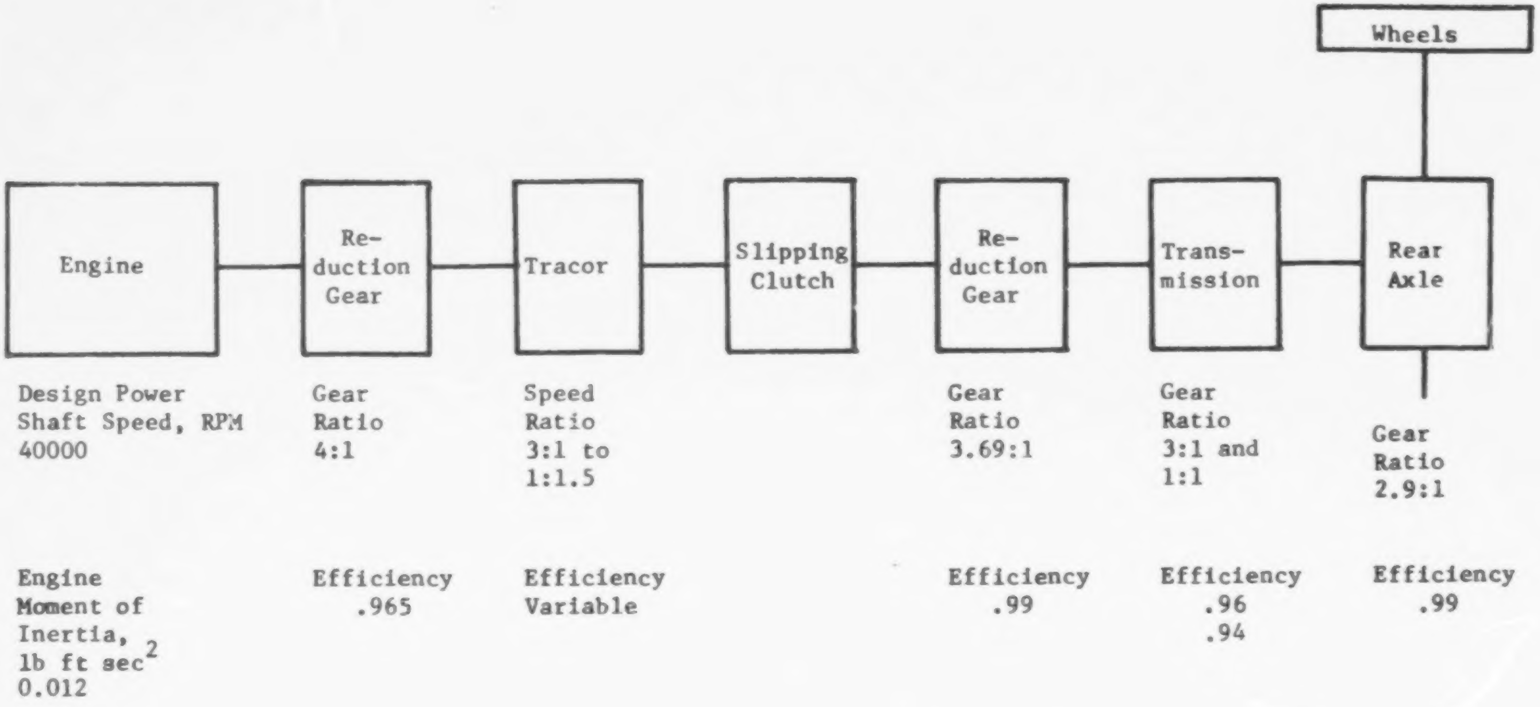

Wheels
Engine
Re-duction Gear
Tracor
Slipping Clutch
Re-duction Gear
Trans-mission
Rear Axle
Design Power Shaft Speed, RPM 40000
Gear Ratio 4:1
Speed Ratio 3:1 to 1:1.5
Gear Ratio 3.69:1
Gear Ratio 3:1 and 1:1
Gear Ratio 2.9:1
Engine Moment of Inertia, lb ft sec^2 0.012
Efficiency .965
Efficiency Variable
Efficiency .99
Efficiency .96 .94
Efficiency .99

Figure 85. Schematic of Drive Train for PD-1A(Single-Shaft) Engine

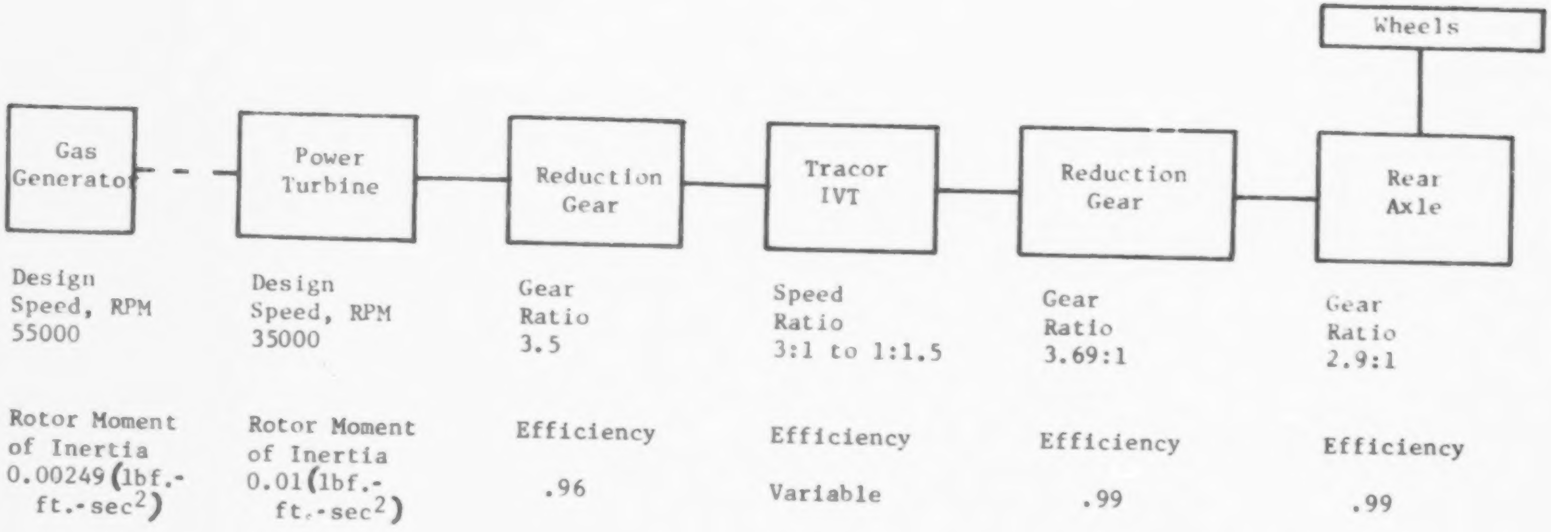

Wheels
Gas Generator
Power Turbine
Reduction Gear
Tracor IVT
Reduction Gear
Rear Axle
Design Speed, RPM 55000
Design Speed, RPM 35000
Gear Ratio 3.5
Speed Ratio 3:1 to 1:1.5
Gear Ratio 3.69:1
Gear Ratio 2.9:1
Rotor Moment of Inertia 0.00249 (lbf.-ft.-sec2)
Rotor Moment of Inertia 0.01 (lbf.-ft.-sec2)
Efficiency .96
Efficiency Variable
Efficiency .99
Efficiency .99

Figure 86. Schematic of Drive Train for CD-2 (Free-Turbine) Engine

The engine transient analysis described earlier in Section 3.6.1 was used to calculate the WOT acceleration performance of engine CD-2.

PD-2A Free Turbine Engine - Figure 87 is a schematic of the drive train of the **PD-2A** (Free Turbine) engine. The power turbine of the PD-2A engine drives the accessories and has an idle speed 16% of design speed and the use of a torque converter permits zero vehicle velocity with non-zero power turbine speed. The characteristics of the torque converter sized for engine **PD-2A** were given previously in Figures 9 and 10 ; the match between this torque converter and the **PD-2A** engine was shown in Figure 11.

A 3-speed automatic transmission is also part of the drive train. The shift points of this transmission are given in Figure 88 ; they were selected as being the points at which the torque at the wheels through the gears is at a maximum for a given fuel flow (or throttle setting).

For the Federal Driving Cycle, the speed ratio across the torque converter is obtained from the torque converter characteristics. Using the torque converter characteristics of Figures 9 and 10 it can be seen from Figure 89 that the following relationship linear in speed ratio in two regions exists

$$\left[\frac{T_O}{T_I}\ \frac{T_I}{{N_I}^2}\right] = a + b\left(\frac{N_O}{N_I}\right) \tag{21}$$

where N and T are speed and torque and the subscripts I and O refer to input and output values. The symbols a and b are constants having the following values in the two regions of Figure 89.

	$0 \leq (N_O/N_I) \leq 0.90$	$0.9 \leq (N_O/N_I) \leq 0.99$
a	0.7735×10^{4}	1.25×10^{-4}
b	-0.7206×10^{-4}	-1.25×10^{-4}

By substituting Equation 21 into the following

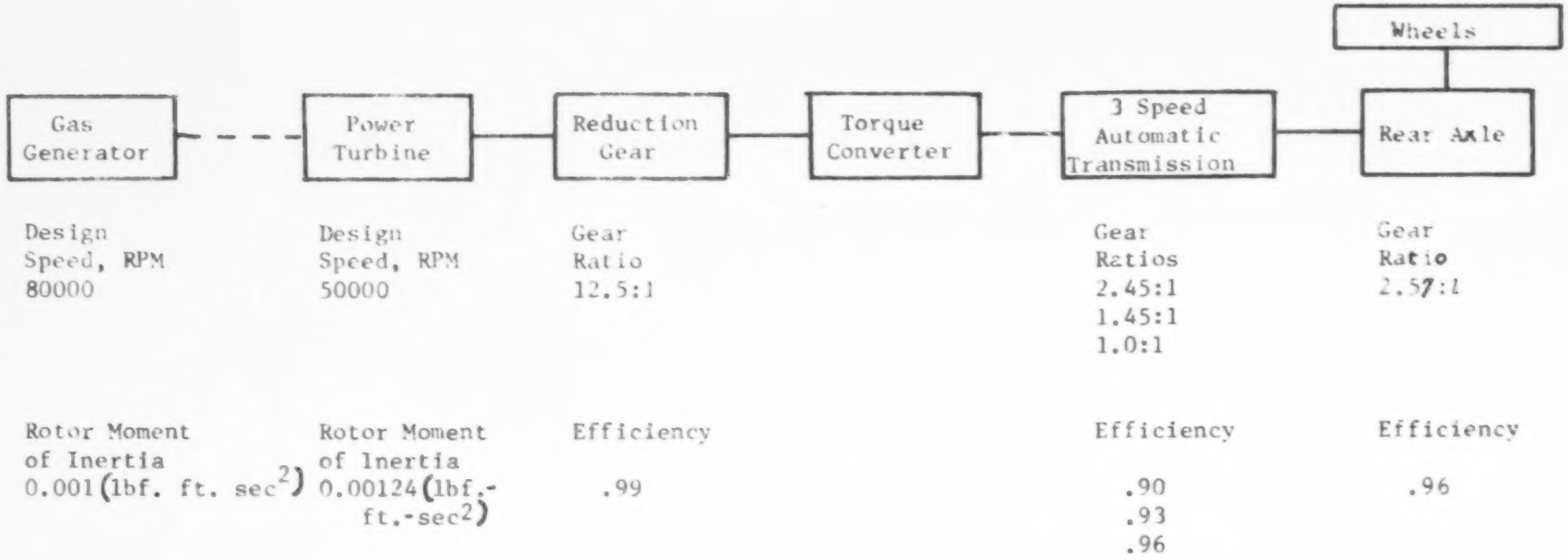

Gas Generator
Power Turbine
Reduction Gear
Torque Converter
3 Speed Automatic Transmission
Rear Axle
Wheels
Design Speed, RPM 80000
Design Speed, RPM 50000
Gear Ratio 12.5:1
Gear Ratios 2.45:1 1.45:1 1.0:1
Gear Ratio 2.57:1
Rotor Moment of Inertia 0.001(lbf. ft. sec2)
Rotor Moment of Inertia 0.00124(lbf.-ft.-sec2)
Efficiency .99
Efficiency .90 .93 .96
Efficiency .96

Figure 87. Schematic of Drive Train for PD-2A(Free-Turbine) Engine

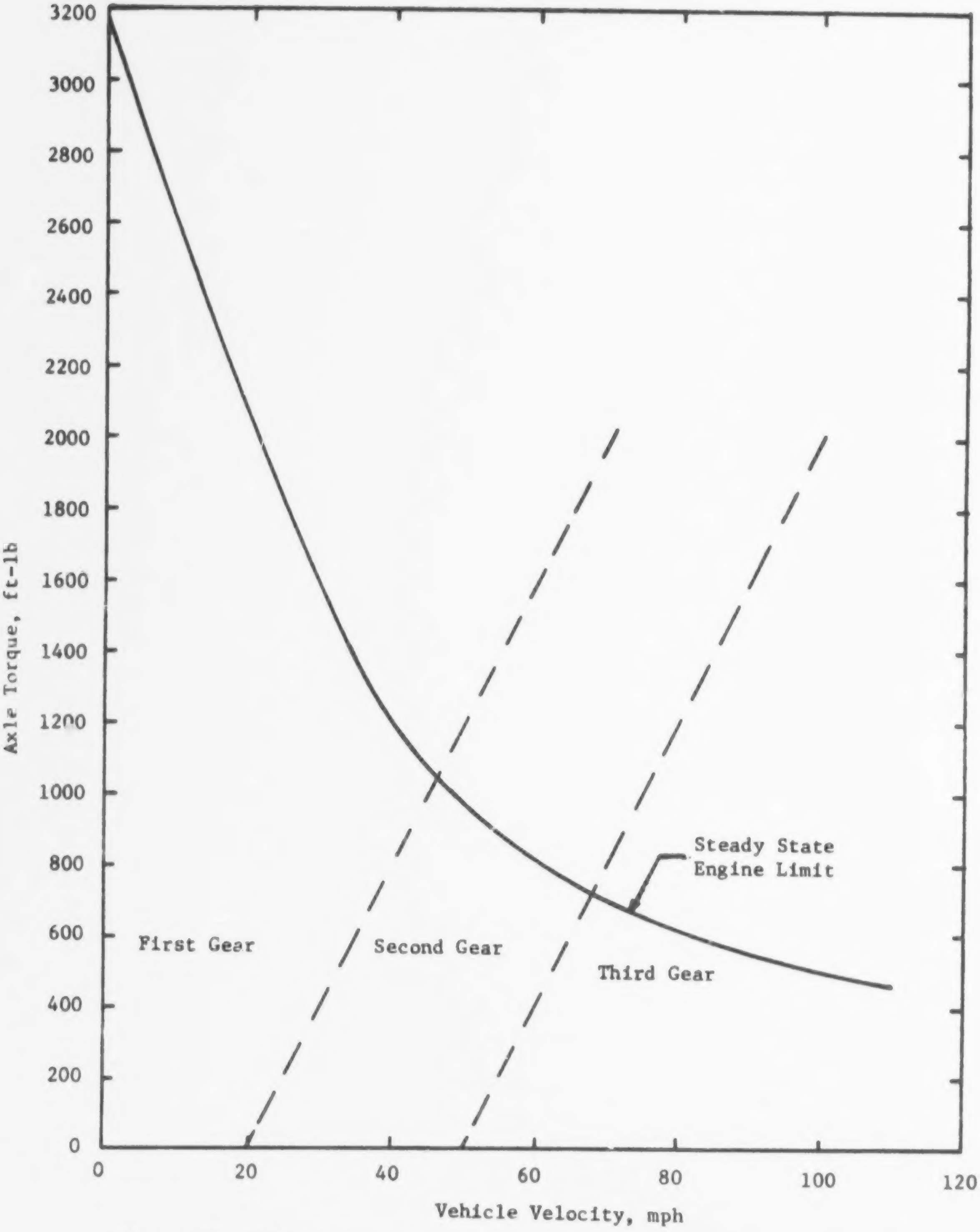


Figure 88. Shift Points for 3-Speed Automatic Transmission for PD-2A (Free-Turbine) Engine

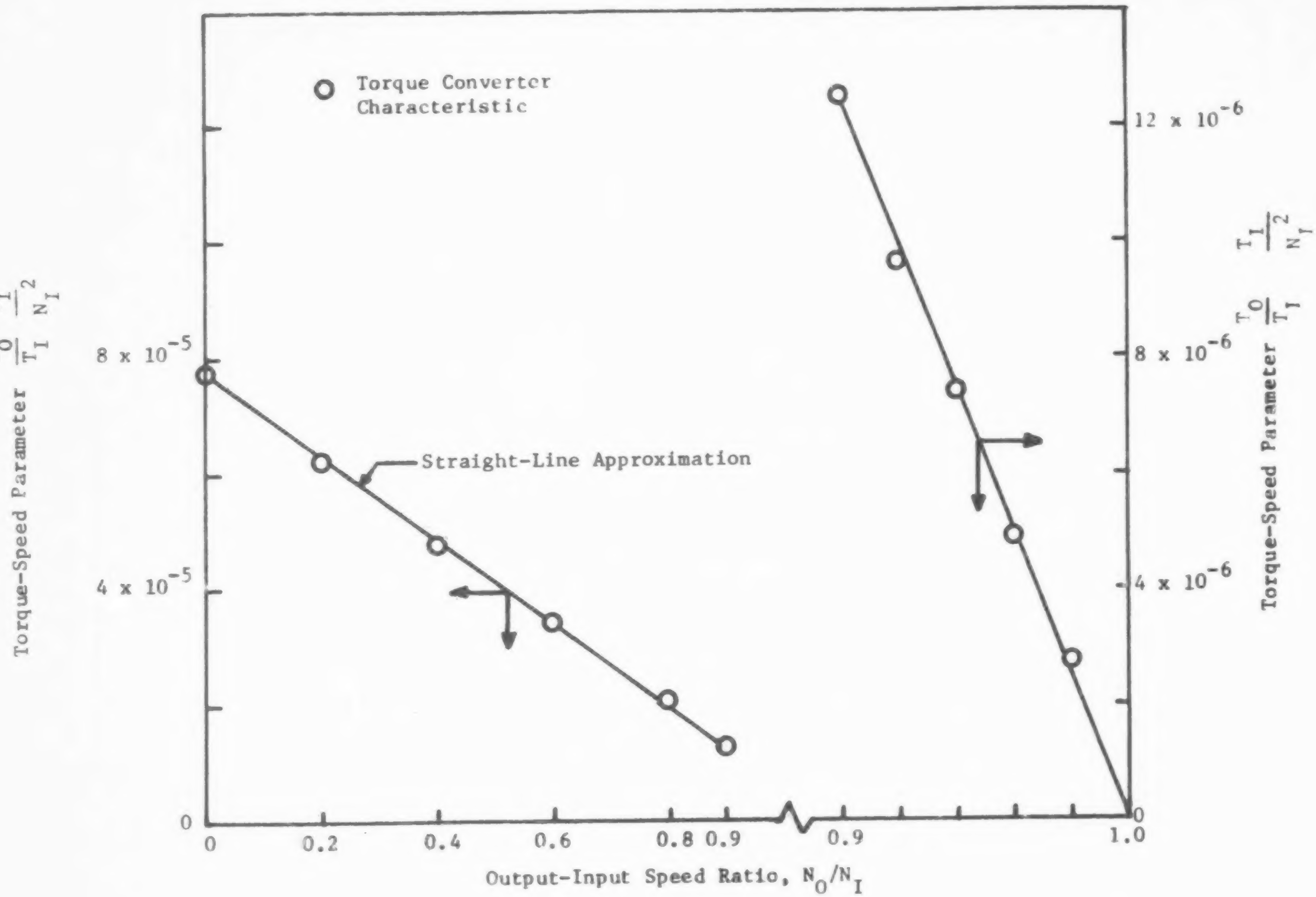


Figure 89. Derived Torque Converter Characteristic Used on Engine PD-2A

$$\frac{N_O^{\,2}}{T_O} = \frac{T_I}{T_O}\,\frac{N_I^{\,2}}{T_I}\left(\frac{N_O}{N_I}\right)^2 \qquad (22)$$

a quadratic equation in (N_O/N_I) results

$$\left(\frac{N_O}{N_I}\right)^2 - b\left(\frac{N_O^{\,2}}{T_O}\right)\left(\frac{N_O}{N_I}\right) - a\left(\frac{N_O^{\,2}}{T_O}\right) = 0 \qquad (23)$$

Shown in Figure 90 is the solution of Equation 23 . For the Federal Driving Cycle the torque converter output speed and torque can be calculated from the current values of the vehicle velocity and the tractive effort required by the Federal Driving Cycle. From these values the speed ratio of the torque converter can be obtained from Figure 90 . Knowing the torque converter speed ratio makes it possible to determine the input torque from Figure 9 and the required engine speed. From these quantities,the total required engine torque can be determined along with the fuel flow rate.

For the WOT acceleration performance of engine PD-2A,the engine transient analysis described earlier in Section 3.6.1 was employed.

Internal Combustion Engine - The schematic of a drive train for an internal combustion engine (176 HP) is given in Figure 91. A torque converter, the characteristics of which are given in Figures 92 and 93 was sized to match this engine. A 3-speed automatic transmission is also used, and the shift points are given in Figure 94. The fuel rate-speed map for this engine is given in Figure 95 and the engine idle was taken as the lowest speed shown in the map (800 RPM).

The procedure for calculating the Federal Driving Cycle vehicle performance is the same as set forth for the PD-2A engine. For wide-open-throttle acceleration, the torque converter input and output speeds can be determined from the current engine speed and vehicle velocity, respectively. Then, the maximum output torque available for vehicle acceleration can be obtained from Equation 21. This approach was also used for the DOT passing maneuver and the merging maneuver; the vehicle

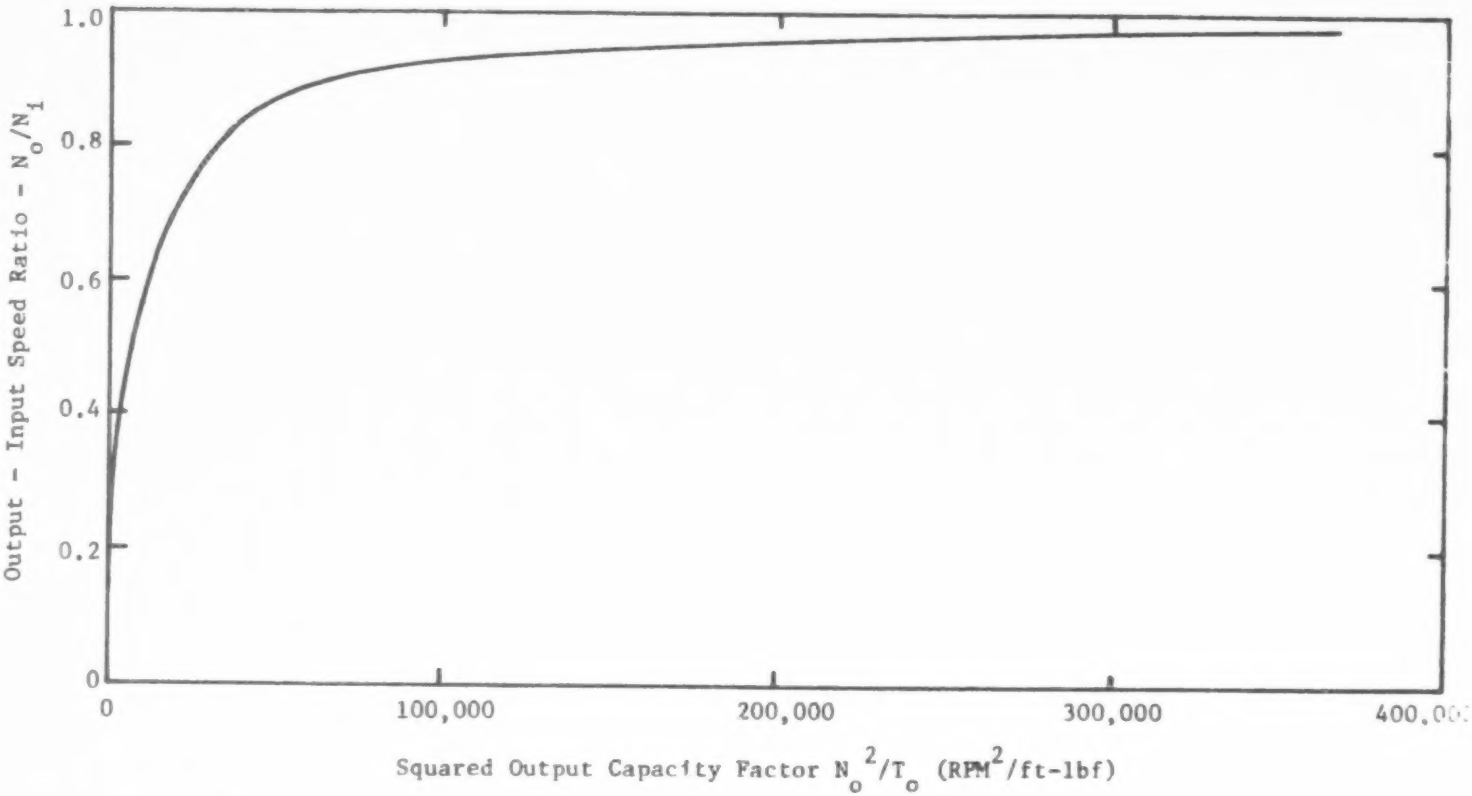


Figure 90. Torque Converter Speed Ratio versus Output Capacity Factor Squared (Engine PD-2A

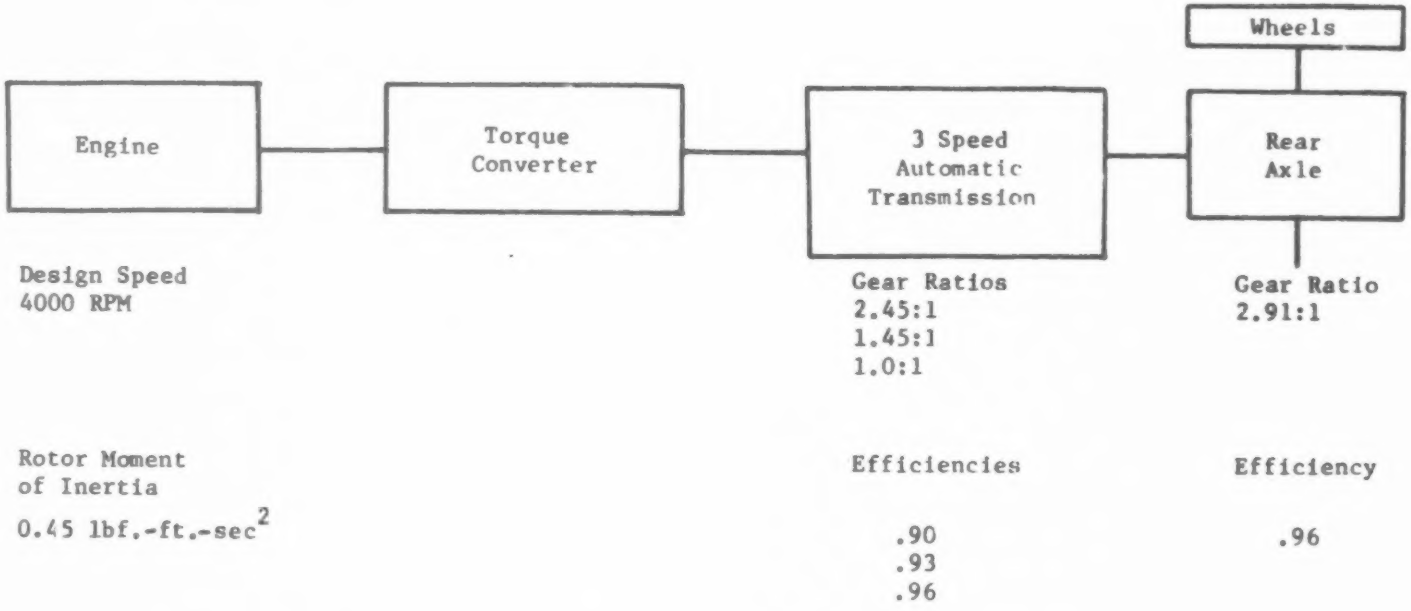

Wheels
Engine
Torque Converter
3 Speed Automatic Transmission
Rear Axle
Design Speed 4000 RPM
Rotor Moment of Inertia
0.45 lbf.-ft.-sec^2
Gear Ratios
2.45:1
1.45:1
1.0:1
Efficiencies
.90
.93
.96
Gear Ratio
2.91:1
Efficiency
.96

Figure 91. Schematic of Drive Train for Internal Combustion Engine

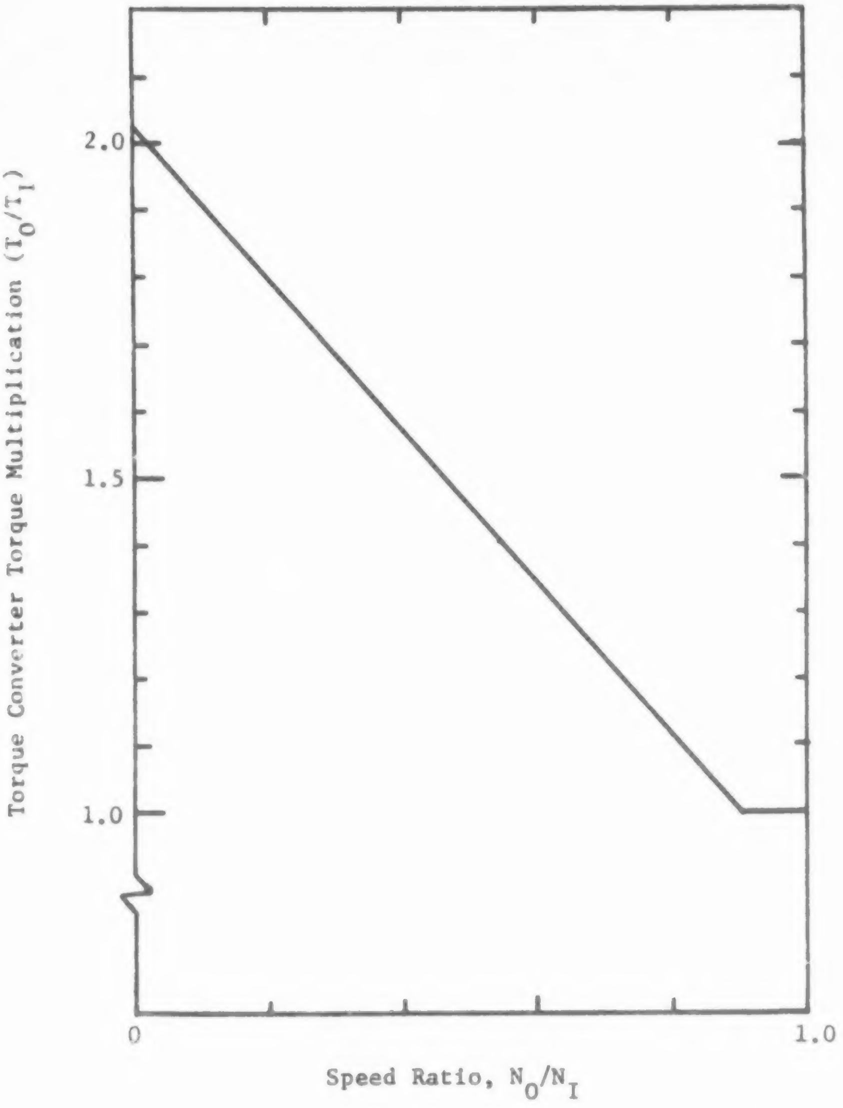


Figure 92. Torque Converter Torque Multiplication

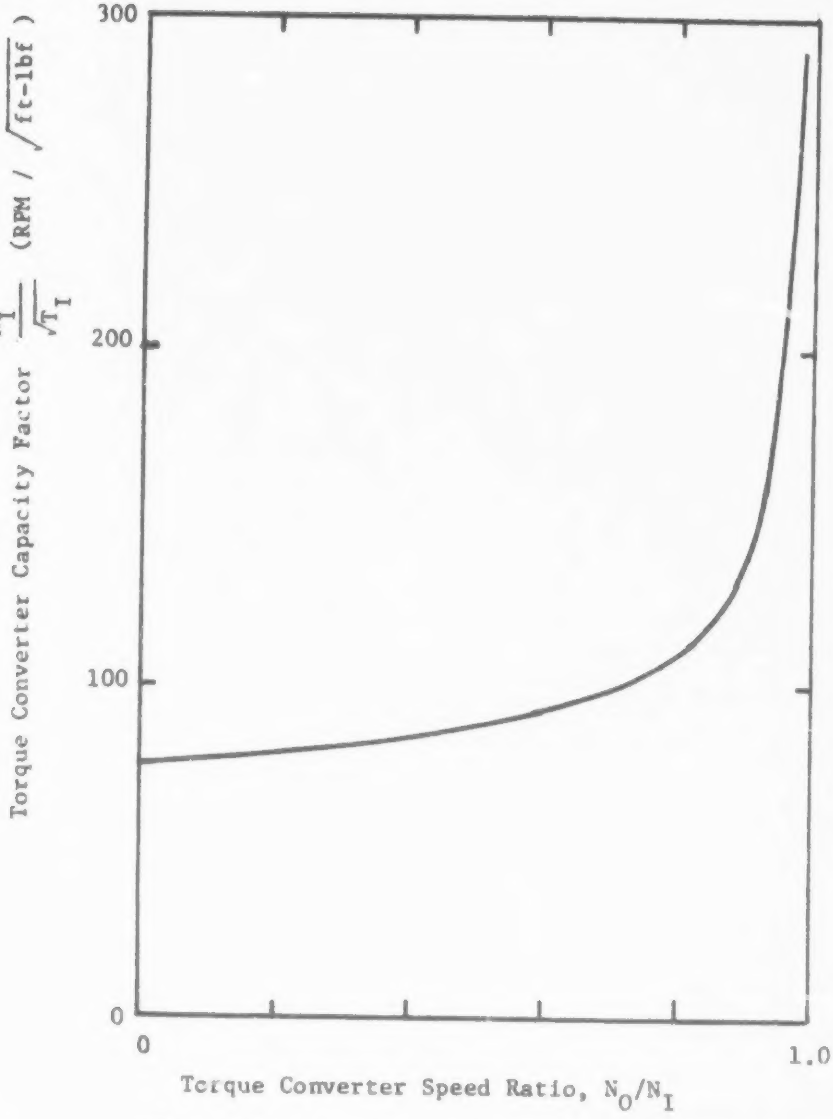


Figure 93. Capacity Factor of Torque Converter for Internal Combustion Engine

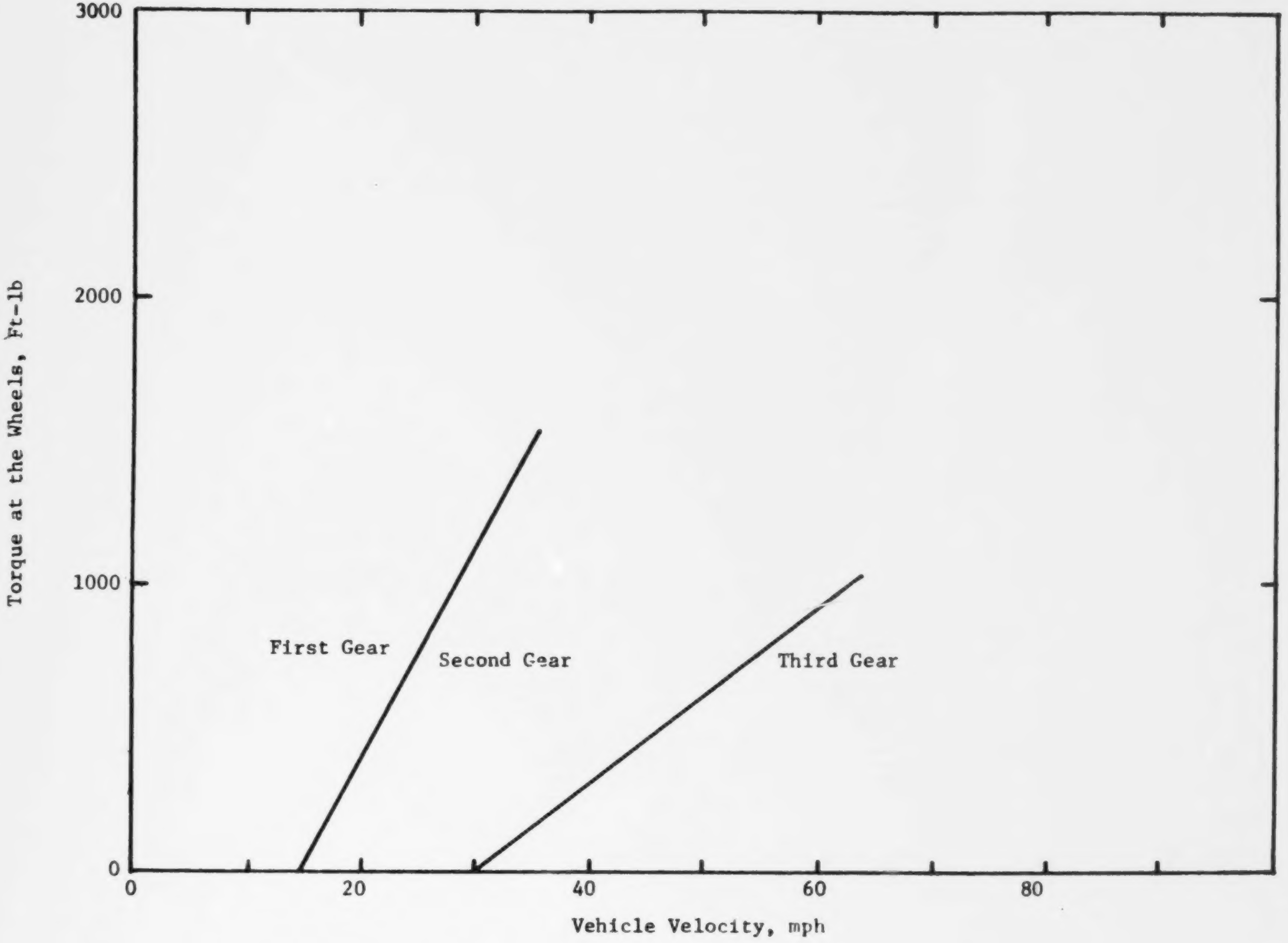


Figure 94. Shifts Points of 3-Speed Automatic Transmission Selected for Internal Combustion Engine

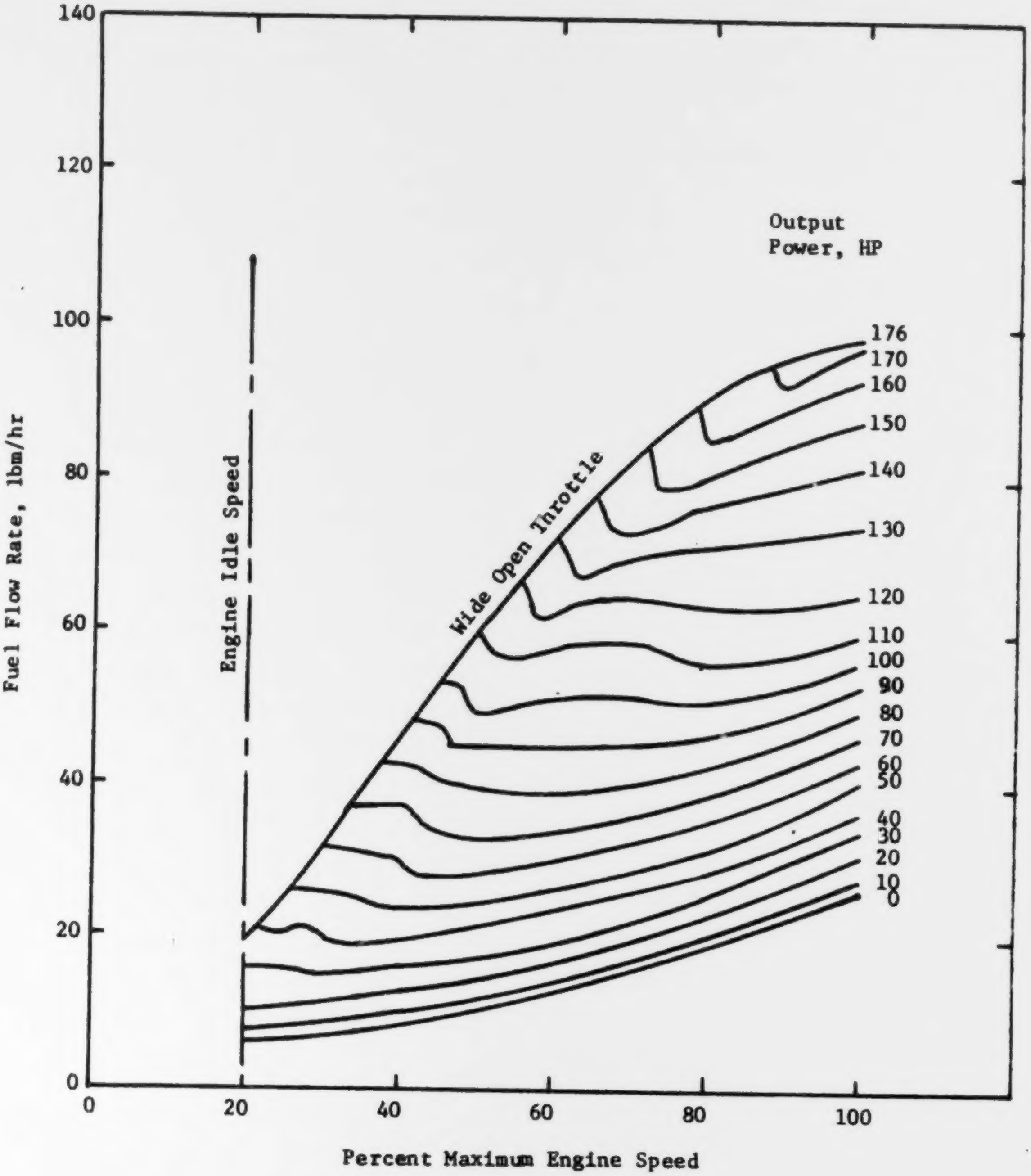


Figure 95. Internal Combustion Engine Performance

cruise velocity was used, and the initial engine speed was set from the torque converter characteristic to provide the necessary output torque and wheel speed at the cruise condition.

3.6.3 Results

The results of the Mission Analyses include engine-vehicle acceleration performance and the total fuel consumption during the life of the automobile.

Engine-Vehicle Acceleration Performance - The results of the acceleration calculations are given in Table 20. For the free turbine engines, the calculations were done using a transient thermodynamic model described earlier. Because of the freedom of the gas generator rotor this more sophisticated technique is from the power train required to properly account for the gas generator inertia. For the PD-1A single shaft engine, the acceleration calculations were done two ways, using the transient thermodynamic model and using a model based on steady state performance data. The transient model requires considerably more input information, such as the specification of a control system and the results of transient stator schedule analyses. The direct coupling of the engine to the drive train permits the use of steady-state code which is adequate for conceptual design studies. The results of the two methods were not much different. The transient model gave slightly slower accelerations due to the assumption of a half second ramp. Shown in Figure 96 are engine speed, tractive effort, vehicle acceleration, distance and velocity as a function of time for the PD-1A WOT acceleration, using the model based on steady state performance data. These results are typical of those obtained from the code.

The corresponding acceleration results for the internal combustion engine are also given in Table 20. The calculational procedure was previously outlined. Initial conditions of engine speed for each of the maneuvers were selected from the torque converter characteristics at the output torque corresponding to the steady-state road load thrust.

The results of the performance of engine PD-1A on the DOT truck passing maneuver are shown in Figure 97; the requirements of the maneuver

Table 20

Acceleration Performance

Engine	Power, HP	Standing Acceleration		Merging 25-70 mph sec	DOT Passing Maneuver		30% Grade mph	5% Grade mph	Max. Speed mph
		10 sec, ft	0-60 mph sec		ft	sec			
Specification		418	14.3	15.8	1470	15.8	14.2	69.5	85
CD-1 (1)	134	462	12.3	15.3	1330	14.4	> 29	> 73	106
CD-2 (2)	150	420	11.6	13.2	1220	12.8	35	84	110
PD-1A (1)	134	464	12.2	15.4	1340	14.6	> 30	> 73	106
PD-1A (2)	134	420	13.0	15.2	1370	15.0	(3)	(3)	105
PD-2A (2)	132	439	11.5	13.6	1240	13.1	33	80	106
ICE (1)	176	489	11.4	13.7	1120	11.6	> 33	> 85	110

(1) Based upon steady-state performance code.

(2) Based upon transient thermodynamic code.

(3) Control system for maximum acceleration on level road did not work for grade velocity.

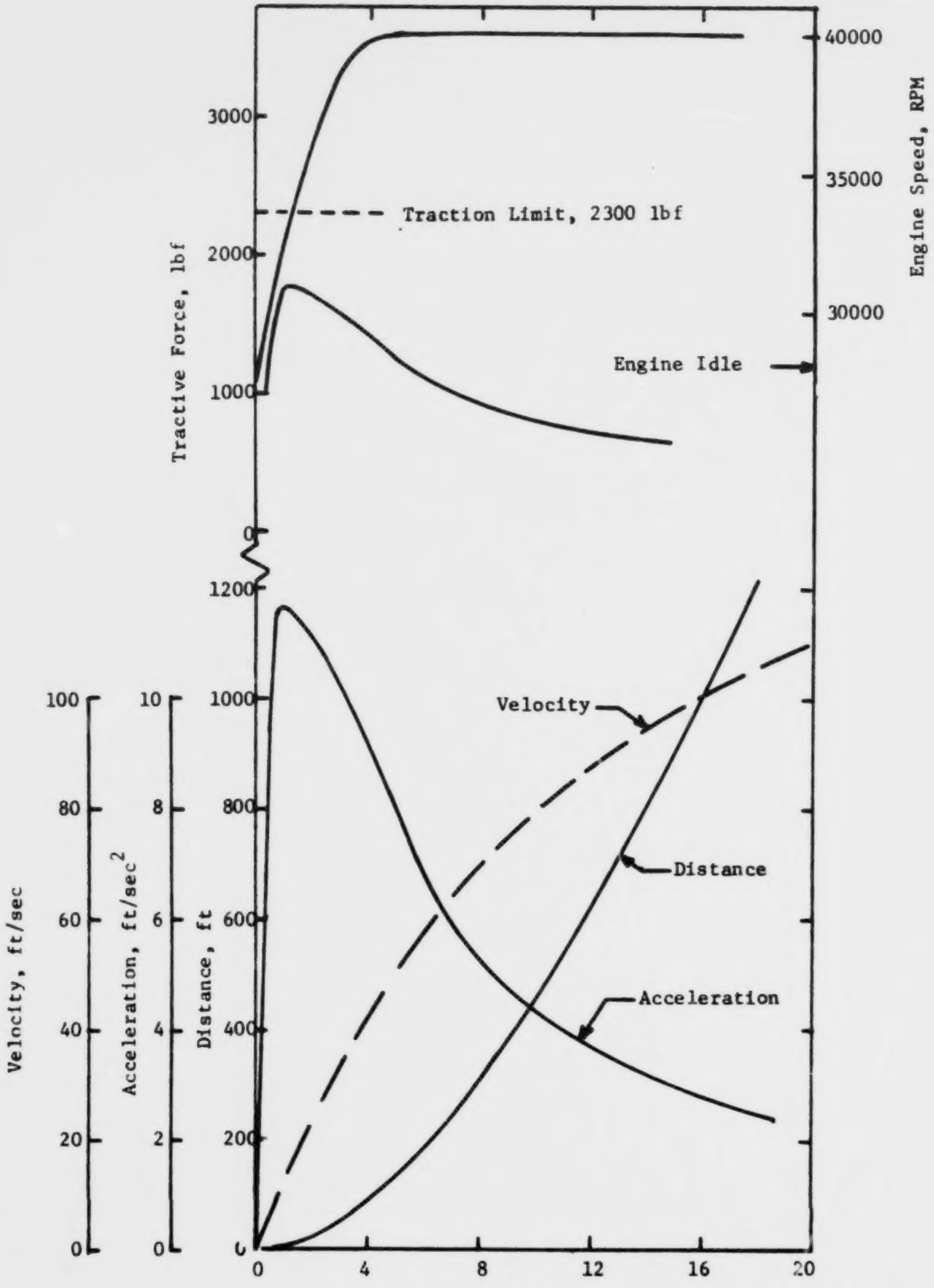


Figure 96. Standing Acceleration of PD-1A Engine (134 HP)

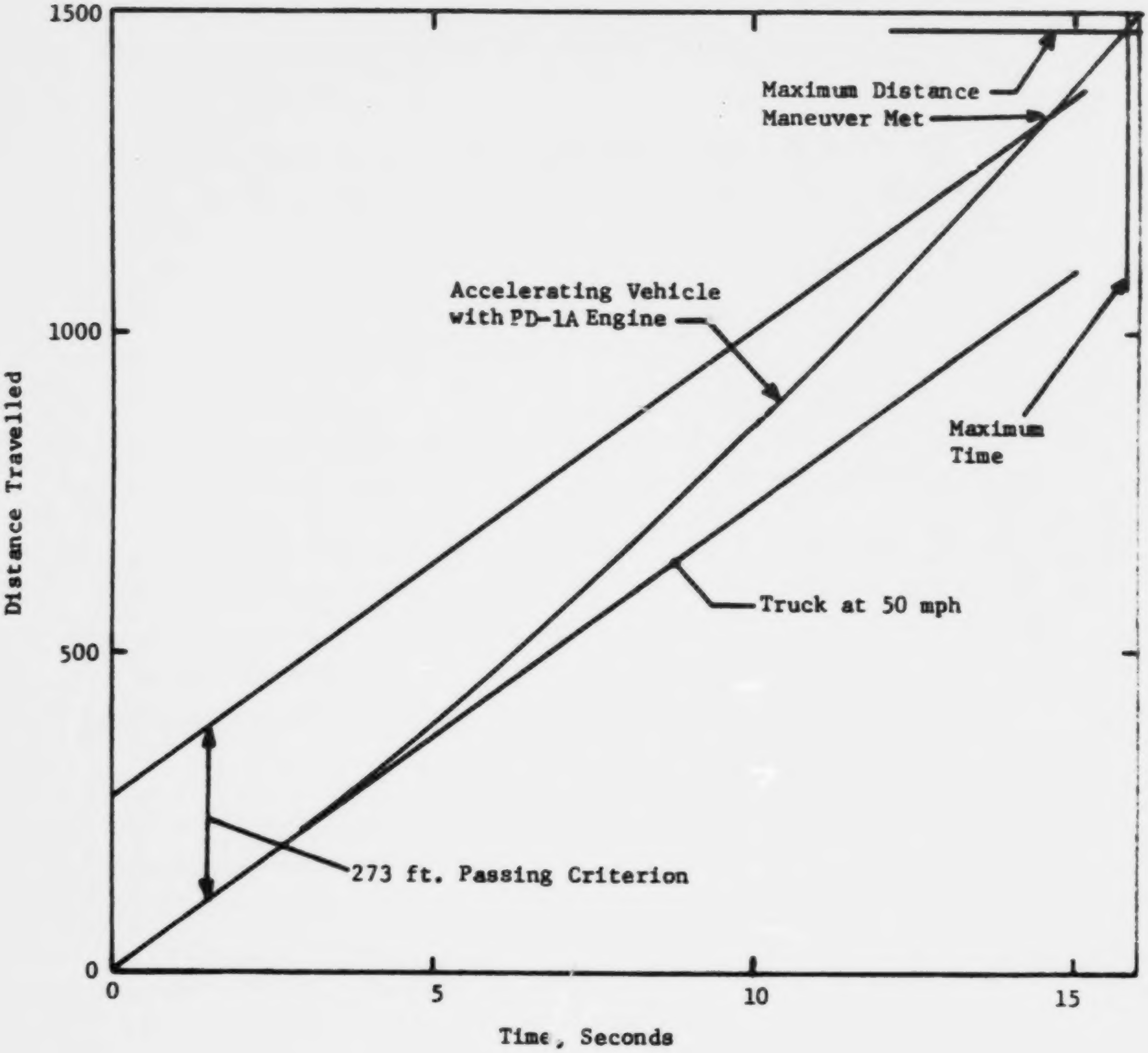


Figure 97. PD-1A Engine Powered Vehicle on DOT Passing Maneuver

are equivalent to having the automobile travel 273 ft. further than the truck. The initial conditions taken for the automobile are a cruise velocity of 50 mph with the engine speed set at that speed corresponding to the cruise horsepower on the line of minimum fuel consumption (70% engine speed).

The effect of engine horsepower (PD-1A engine) upon the distance travelled in 15.75 seconds from an initial condition of 50 mph (steady-state) is:

	$V_* = 53$ mph	$V_* = 60$ mph
120 HP	1421	1342 ft
135 HP	1472	1391 ft

where V_* is the vehicle velocity at which maximum engine speed is attained. The horsepower ratings are on a 105°F day.

By comparison between values of V_*, it is advantageous to put most of the engine power into engine acceleration at the beginning of the maneuver; the reason for this is that as the engine speeds up, more power is available from the engine at its higher speeds. For the case of $V_* = 53$ mph, the engine speed of PD-1A has gone from 70% to 100% within 4 seconds.

As for the 0-60 mph WOT acceleration and the distance travelled in 10 seconds, the following results were obtained for Engine PD-1A (rotor inertia, 0.012 ft-lbf-sec^2) as a function of engine horsepower (105°F day, 4600 lbm vehicle):

	120 HP	135 HP	150 HP	Requirement
Distance Travelled in 10 sec, ft	421	464	504	418
Time to go from 0 to 60 mph, sec	14.1	12.2	10.7	14.2

These results depend on the paradigm chosen for the engine speed versus vehicle velocity; and the one used here has a smooth vehicle acceleration and is the one discussed earlier in the section on acceleration of Single Shaft Engines CD-1 and PD-1A. For these calculations, zero time was allowed for the turbine vanes to open and for the combustion system to supply the required larger flows and higher temperatures.

The effect of rotor inertia of PD-1 engine on wide-open-throttle acceleration was examined; at the time of the calculations, the horsepower was set at 150 horsepower with 70% engine idle, and the two rotors used in this calculation were: 40,000 RPM rotative speed with 0.014 ft-lbf-sec^2 moment of inertia, and 32,750 RPM and 0.028 ft-lbf-sec^2. The calculations of distances travelled were:

	40,000 RPM 0.014 ft-lbf-sec^2	32,750 RPM 0.028 ft-lbf-sec^2
Distance Travelled in:		
1 sec	5.1	4.3
5 sec	140	125
10 sec	497	471

Hence, the 40,000 RPM rotor did improve the acceleration performance of the engine.

FDC Results - Shown in Table 21 is a summary of the fuel economy calculations for the five engines. The techniques utilized for the engines and their drive trains were briefly outlined above; and the techniques and results for the engines from the two EPA contractors are given in Appendix B. The listed horsepowers are rated on a 105°F day.

From Table 21, it is seen that the CD-1 and CD-2 engines have the best fuel economy of the engines studied, with CD-1 having $89 less total fuel cost than CD-2. The internal combustion engine (176 HP as supplied by EPA) has reasonably good fuel economy, only $235 more than CD-1 over the total life of the vehicle. Comparing the two single shaft engines, CD-1 and PD-1A, it is seen that PD-1A has a fuel cost greater by $366. This is the penalty due to the low pollution combustor which has a bypass flow around the regenerator with a resultant penalty in fuel economy. For the PD-2A free turbine engine, low pollution was the reason for limiting the combustor inlet temperature to 1000°F. The resulting fuel penalty was greater than calculated initially, because acceleration requirements led to a lower than optimum pressure ratio of 6.6 with a lower recuperator effectiveness to limit the combustor inlet temperature. For the two low pollution engines, the total fuel costs of the PD-1A are $244 less than those of the PD-2A engine. These fuel economy values for the PD-1A and

Table 21

Fuel Economy

A. FUEL MILEAGE, mpg

Vehicle Velocity, mph

Engine	Design Power HP	FDC	20	30	40	50	60	70	Average
CD-2	150	11.58	15.76	18.95	20.32	20.83	18.43	16.15	15.39
CD-1	134	12.13	14.79	19.76	23.39	21.60	19.35	16.35	16.06
PD-1A	134	10.56	14.35	16.42	18.31	17.23	15.49	13.54	13.59
PD-2A	132	9.27	12.96	15.14	16.23	15.77	14.63	13.42	12.34
Internal Combustion Engine	176	12.17	14.29	15.30	18.61	17.36	15.62	13.82	14.38

B. FUEL COST FOR 105,200 MILES

Vehicle Velocity, mph

Engine	FDC	20	30	40	50	60	70	Total
CD-2	921	151	188	234	149	202	269	2114
CD-1	879	161	181	203	144	192	265	2025
PD-1A	1009	165	217	259	180	240	321	2391
PD-2A	1150	183	236	293	196	254	323	2635
Internal Combustion Engine	876	166	233	255	178	238	314	2260

PD-2A engines were calculated using the modified cycles described under off design performance calculations.

Histograms - The engine loading as a function duration has an effect upon the life of the engine, maintenance and repair frequency. Thus, as each engine was simulated on the Federal Driving Cycle (FDC) the time accumulated at incremental engine operating conditions was recorded. The results are the histograms presented herein. According to the Uniform Simplified Engine Duty Cycle 50% of the engine life is used up by repeating the FDC. The FDC has a duration of 1370 sec. of which 241 sec. represent zero vehicle velocity with the engine running at idle speed.

Shown for all engines studied are the following histograms: engine output speed fraction, engine power fraction, braking power fraction and transmission input-output speed ratio. The fractional values are normalized by the rated conditions. Braking power time is accumulated only when the wheel torque is negative, not merely when the vehicle is decelerated. Because the Tracor transmissions are input torque limited, histograms of transmission input torque are presented for the CD-1, CD-2 and PD-1 engines.

Shown in Figures 98, 99, 100, 101 and 102 are the engine speed fraction histograms. The histograms are similar for the CD-1 and PD-1 engines (Figures 98 and 100) since both are single shaft engines with Tracor transmissions. Most of the time is spent at speeds near the idle speed of 70% rated speed. The histograms for CD-2 and PD-2 (Figures 99 and 101) differ somewhat, though both engines are free-turbine engines; the difference is due to the use of the Tracor transmission on the CD-2 and a conventional automatic transmission on the PD-2 engine. The automatic transmission tends to make the engine run in the mid speed range while the Tracor tends to make the engine run at higher speeds. The histogram (Figure 102) for the internal combustion engine (ICE) is similar to that for the PD-2 engine since both have conventional automatic transmissions.

Shown in Figures 103, 104, 105, 106 and 107 are the engine power fraction histograms and shown in Figures 108, 109, 110, 111, and 112 are the braking power histograms. These histograms are similar for the several

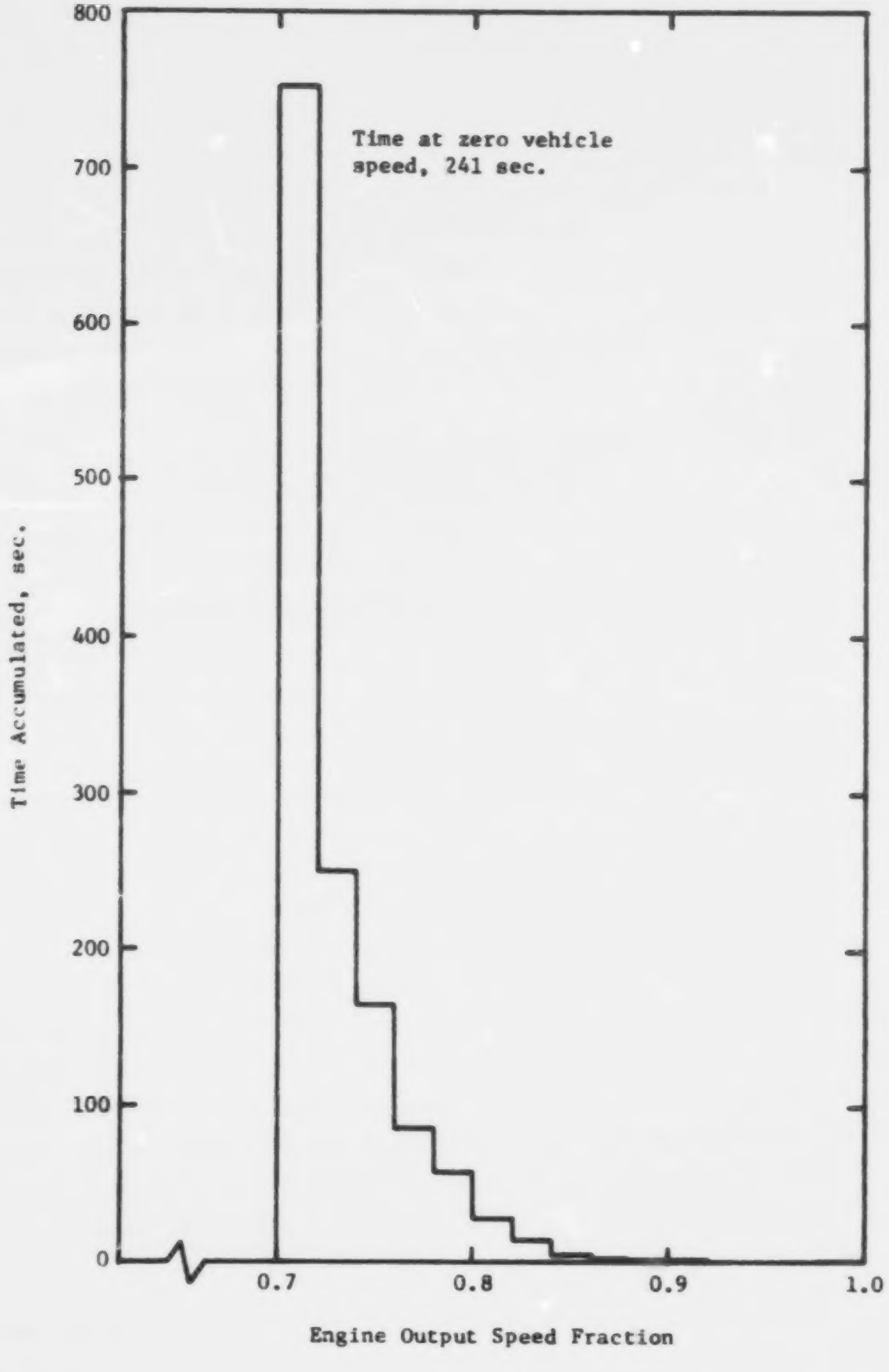
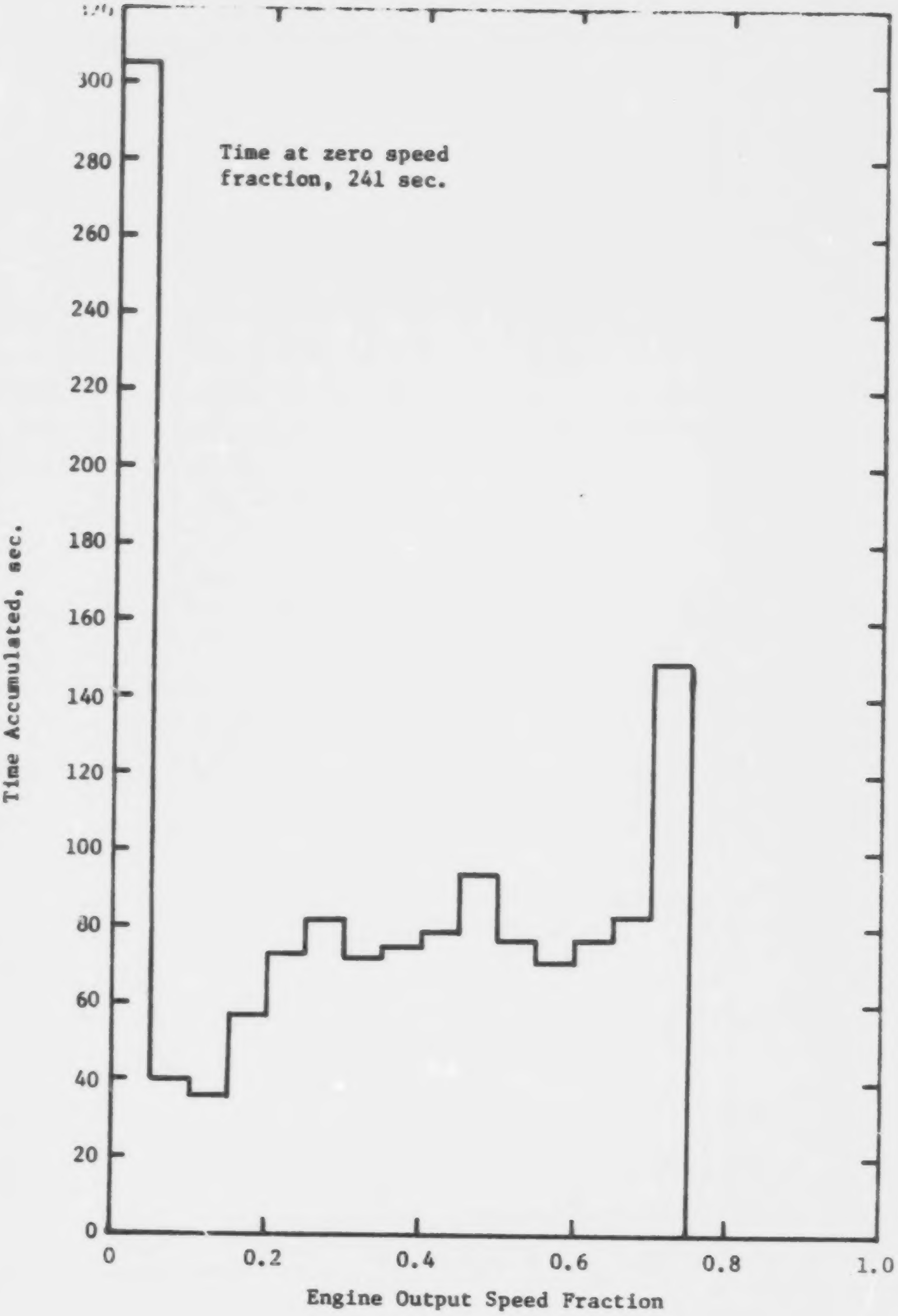


Figure 98. Engine Output Speed Fraction Time Distribution on FDC with CD-1 Engine



Time at zero speed fraction, 241 sec.

Figure 99. Engine Output Speed Fraction Time Distribution on FDC with CD-2 Engine

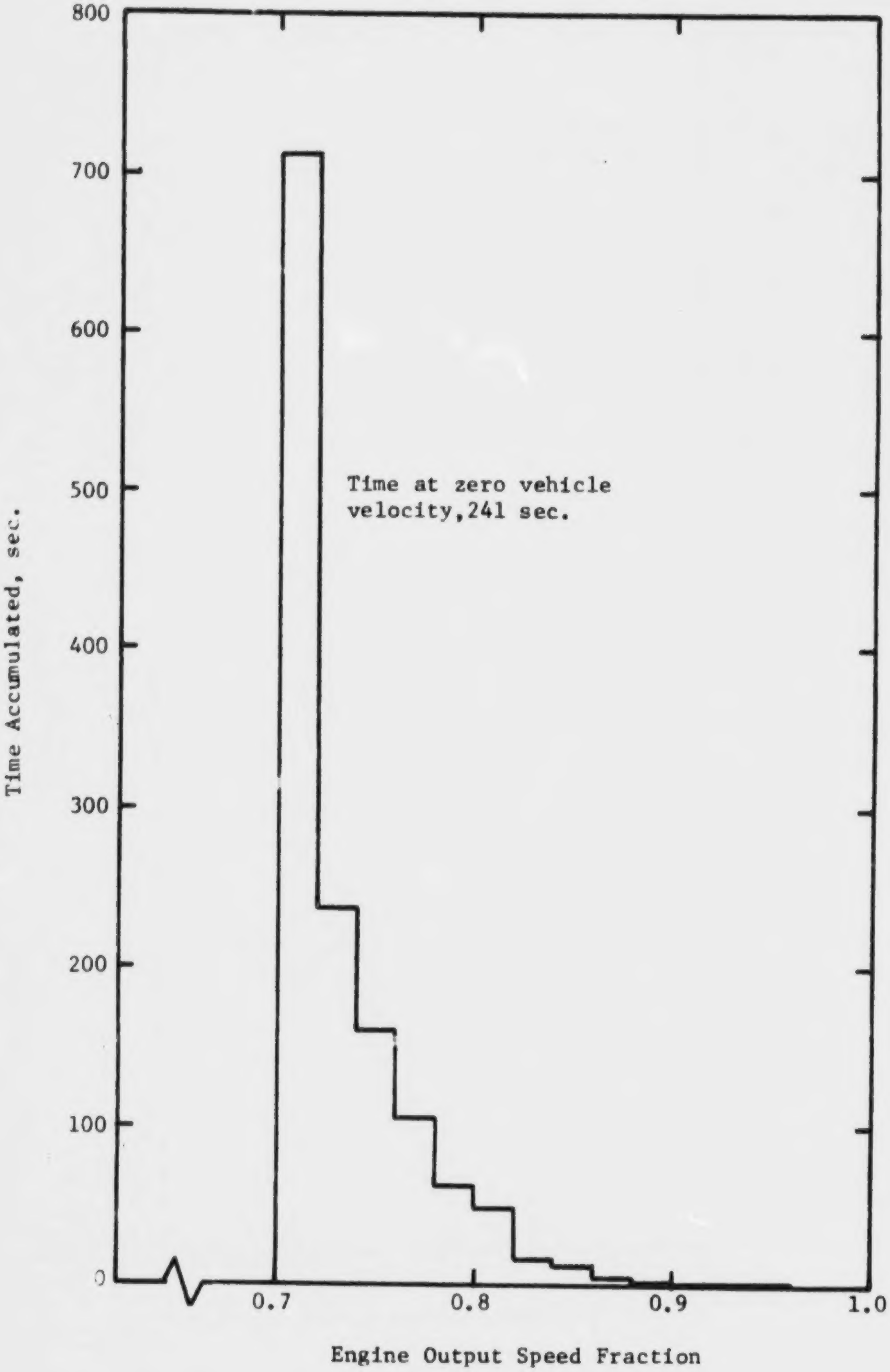


Figure 100. Engine Output Speed Fraction Time Distribution on FDC with PD-1 Engine

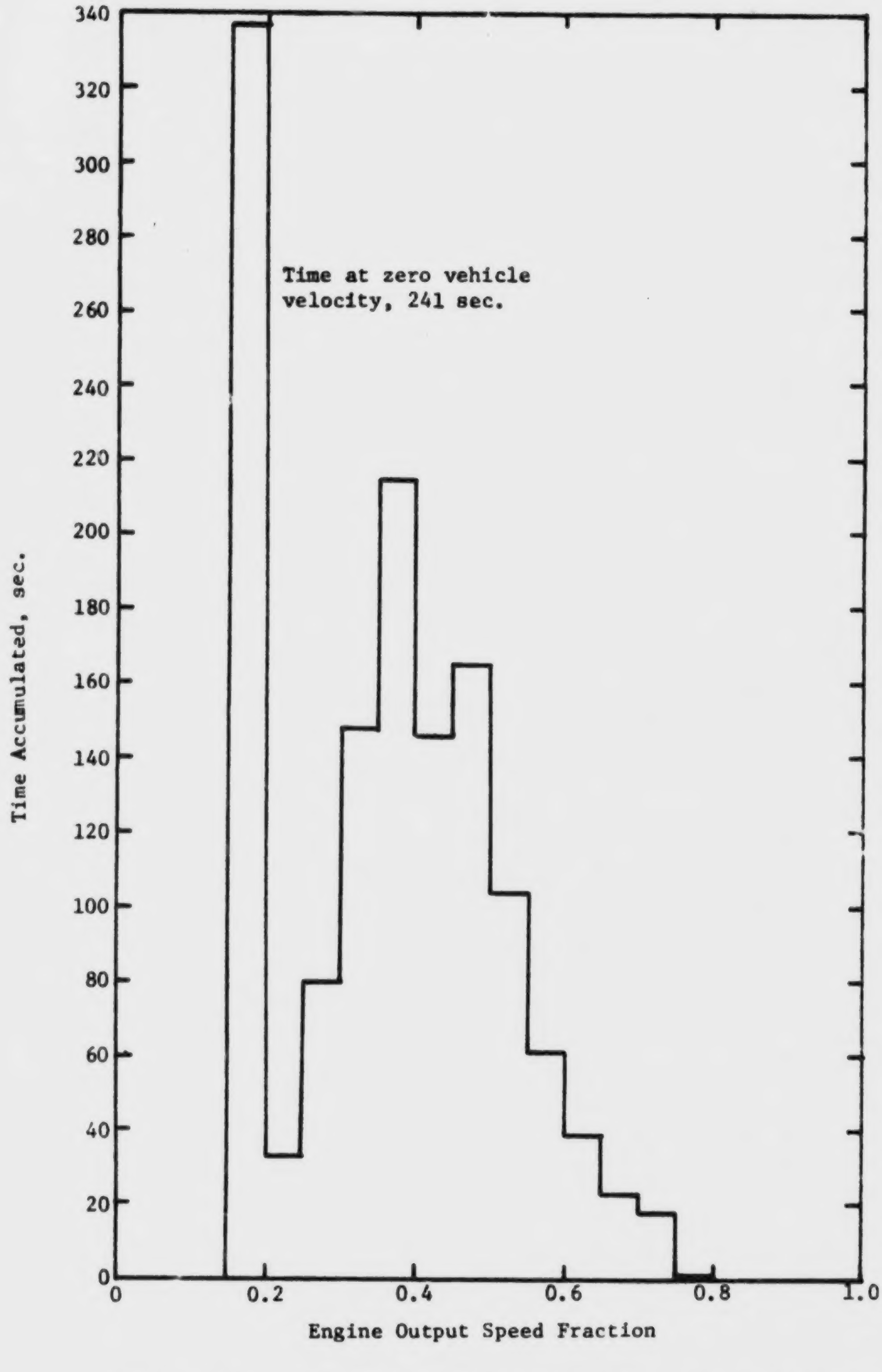


Figure 101. Engine Output Speed Fraction Time Distribution on FDC for PD-2 Engine

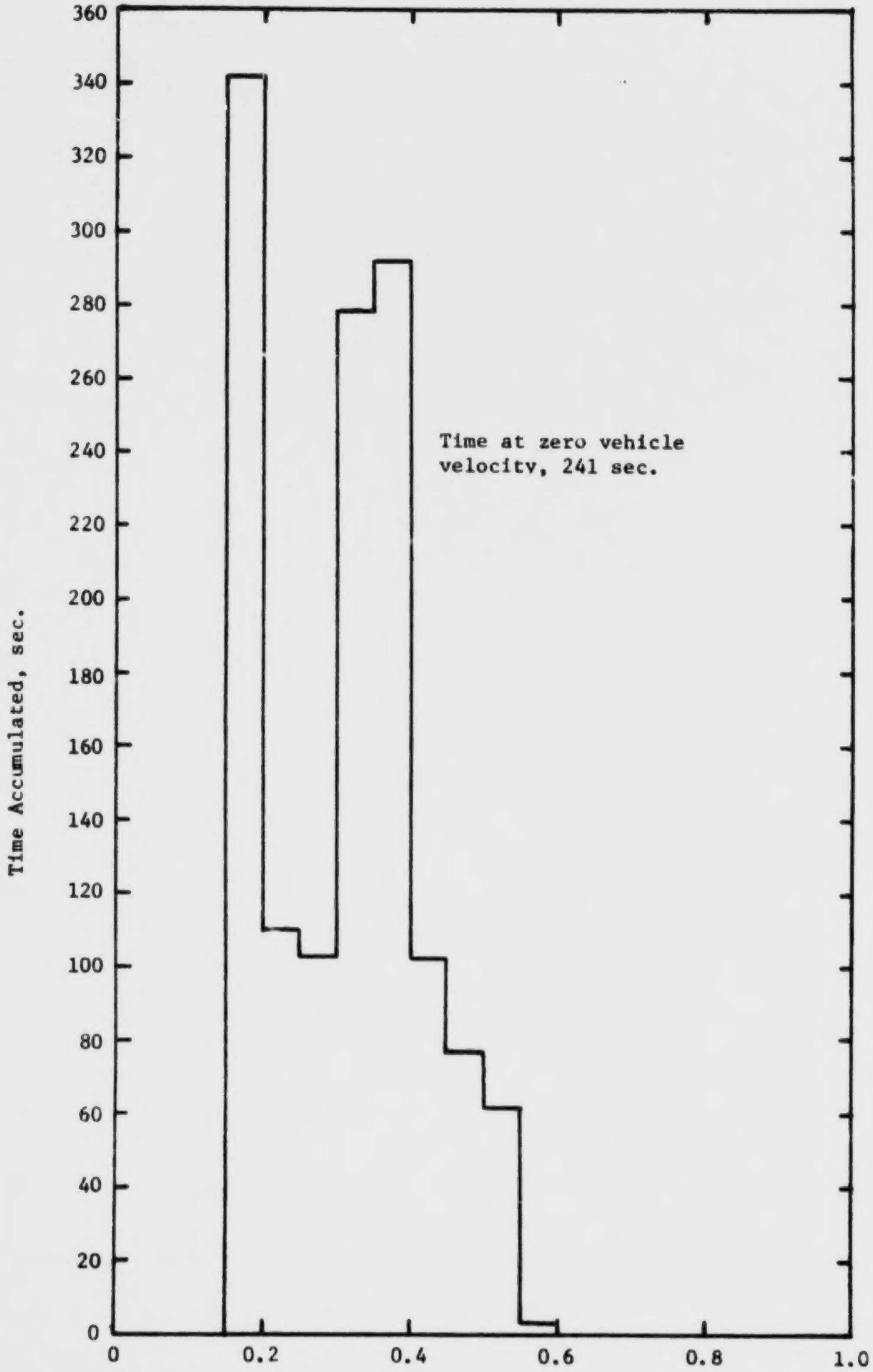


Figure 102. Engine Speed Fraction Time Distribution on FDC for Internal Combustion Engine

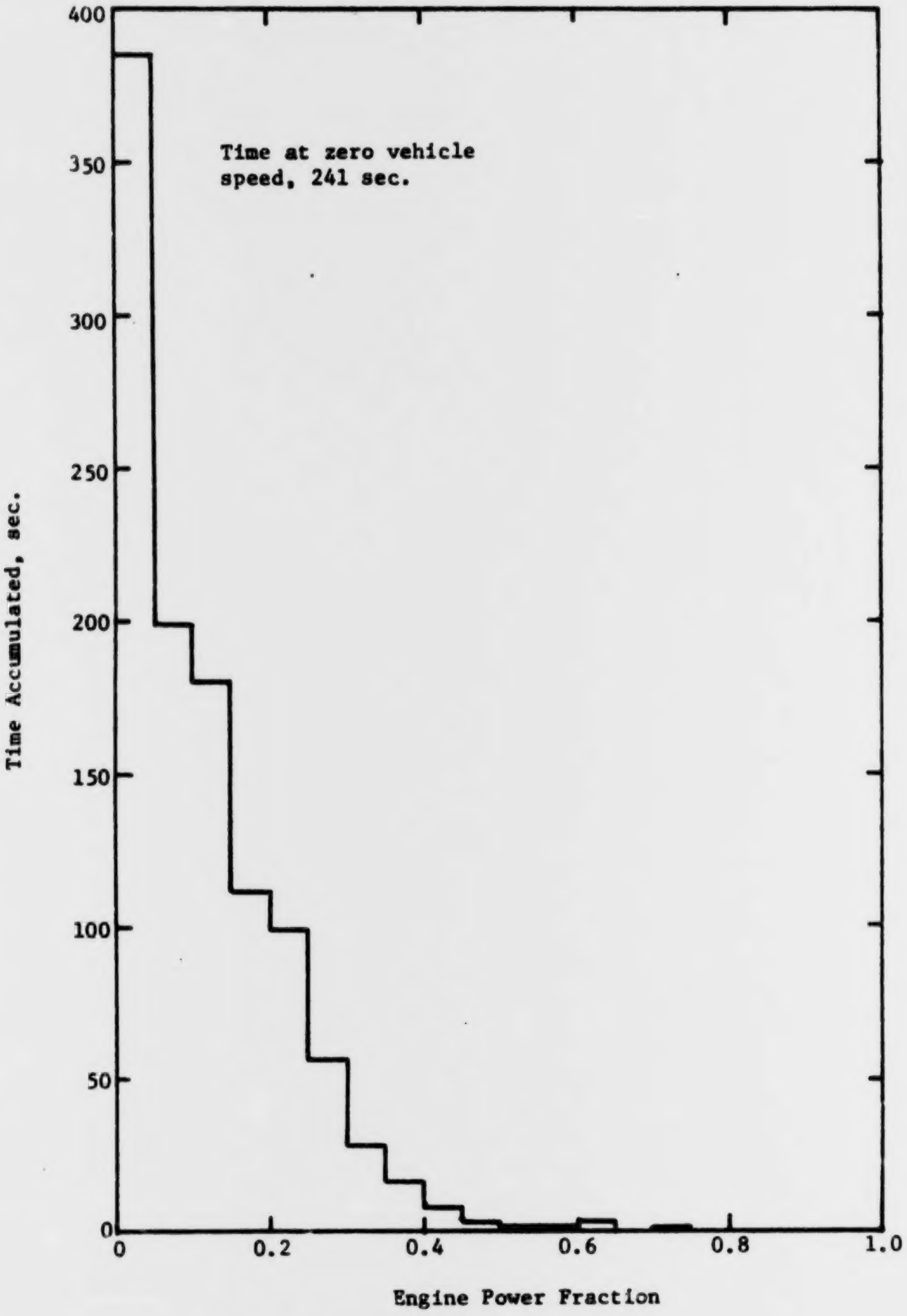


Figure 103. Engine Power Fraction Time Distribution for FDC on the CD-1 Engine

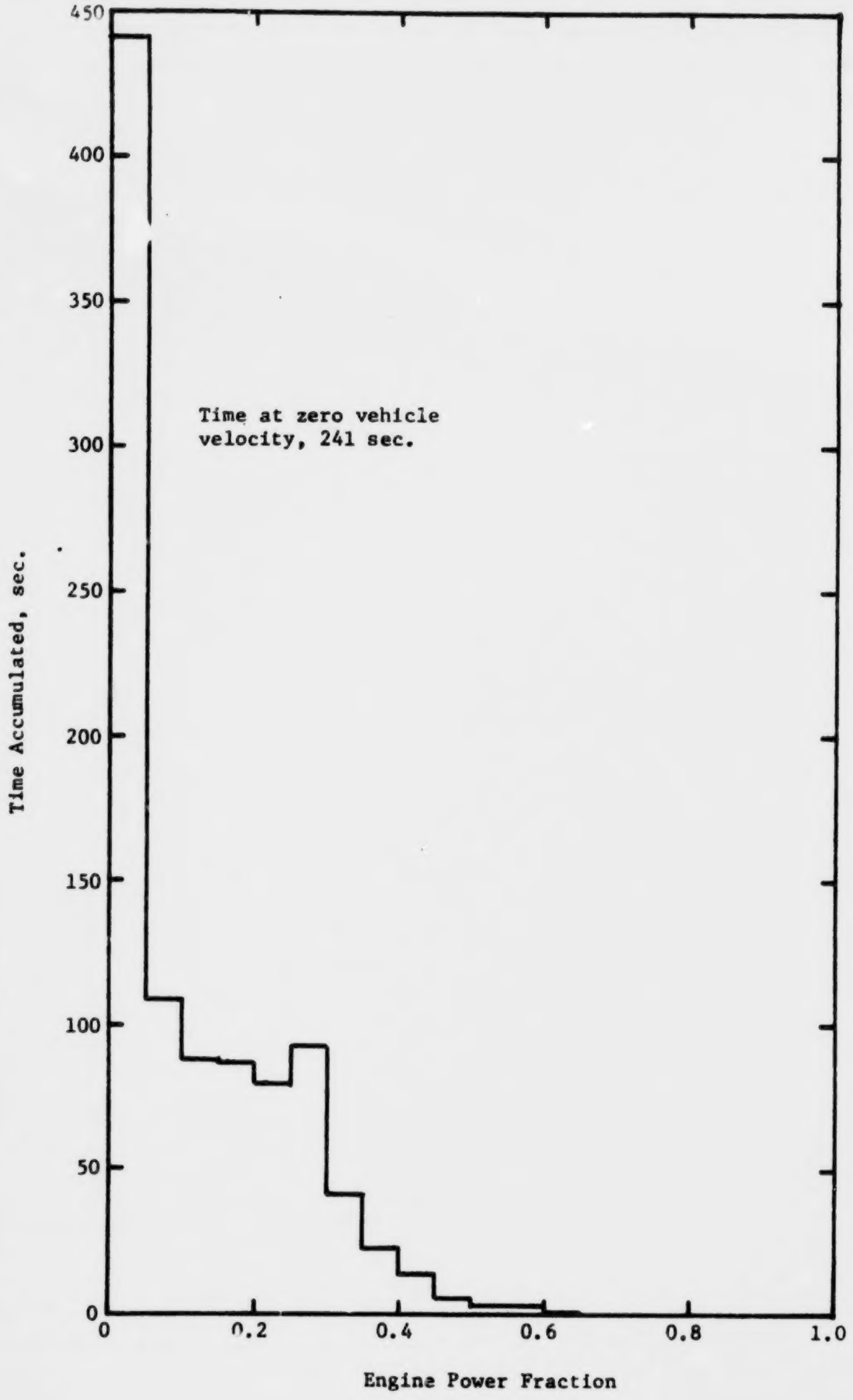


Figure 104. Engine Power Fraction Time Distribution on FDC for CD-2 Engine

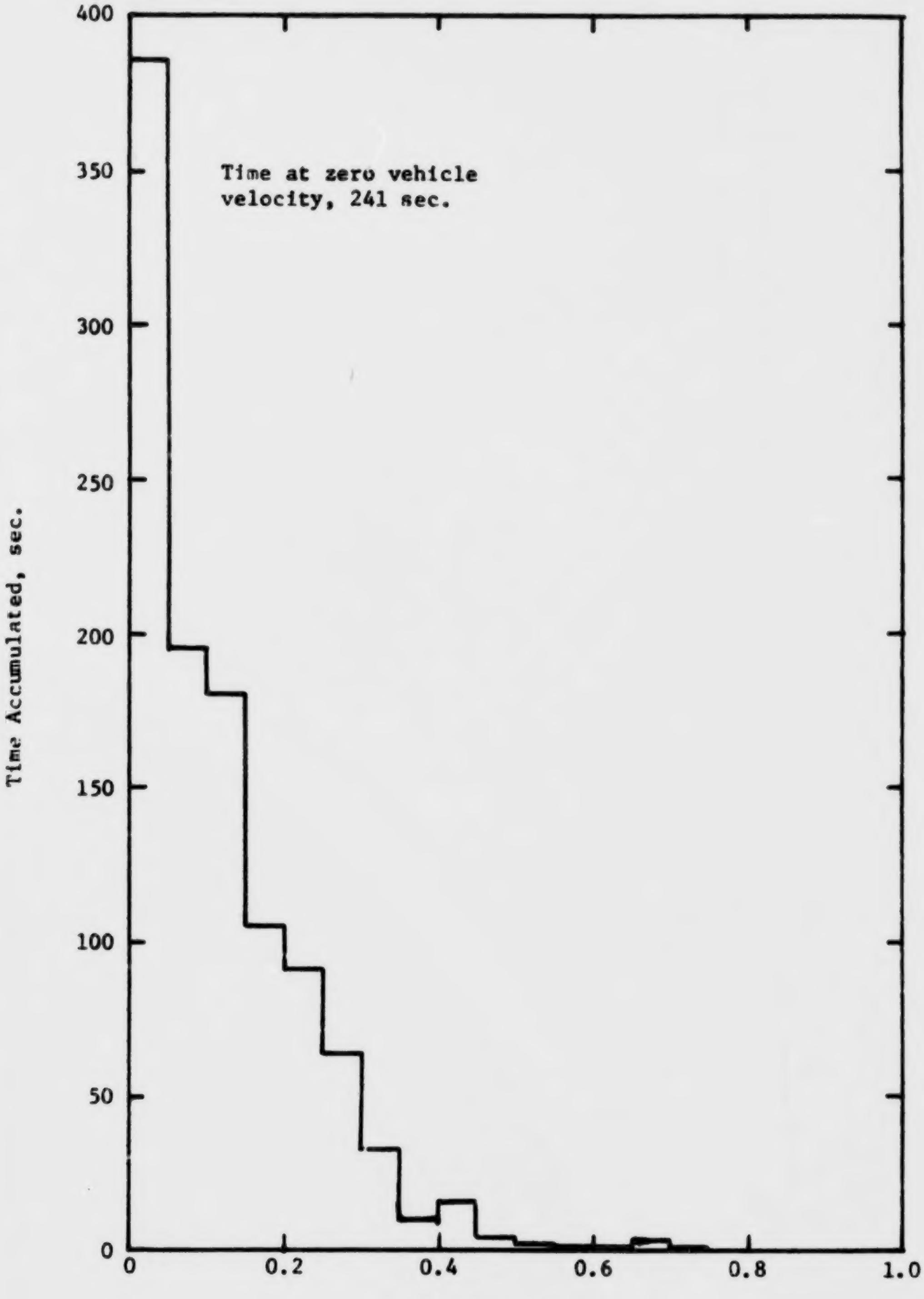


Figure 105. Engine Power Fraction Time Distribution for FDC on the PD-1 Engine

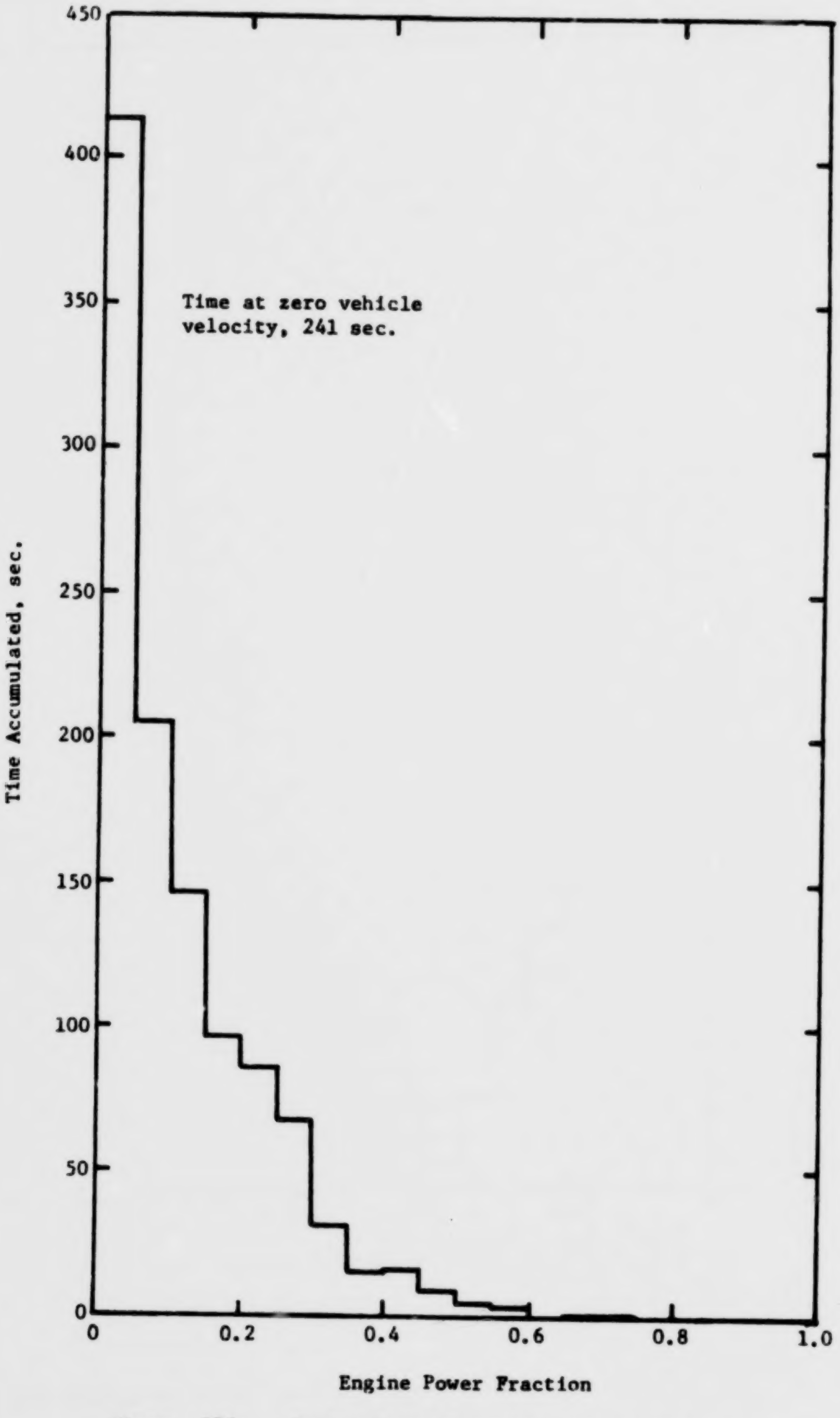


Figure 106. Engine Power Fraction Time Distribution on FDC for PD-2 Engine

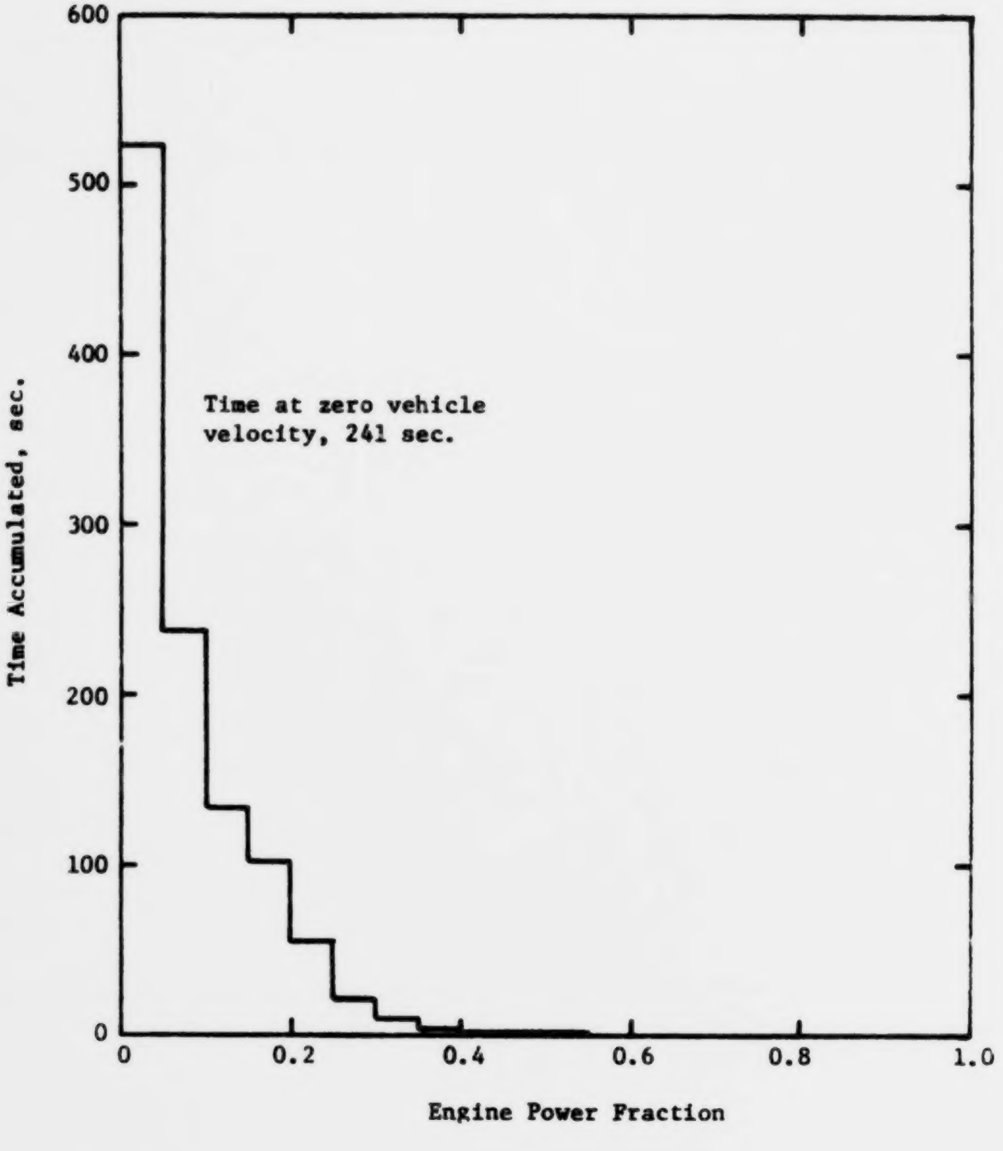


Figure 107. Engine Power Fraction Time Distribution on FDC for Internal Combustion Engine

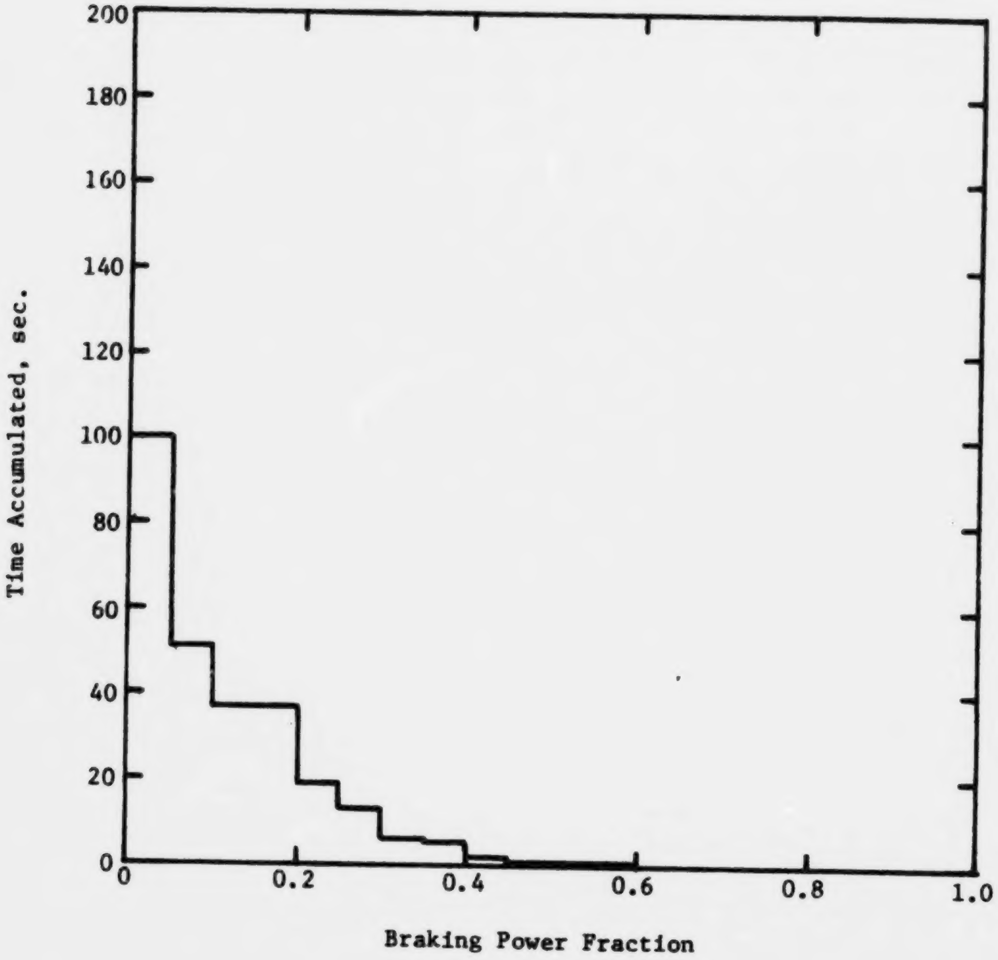


Figure 108. Braking Power Fraction Time Distribution for FDC on the CD-1 Engine

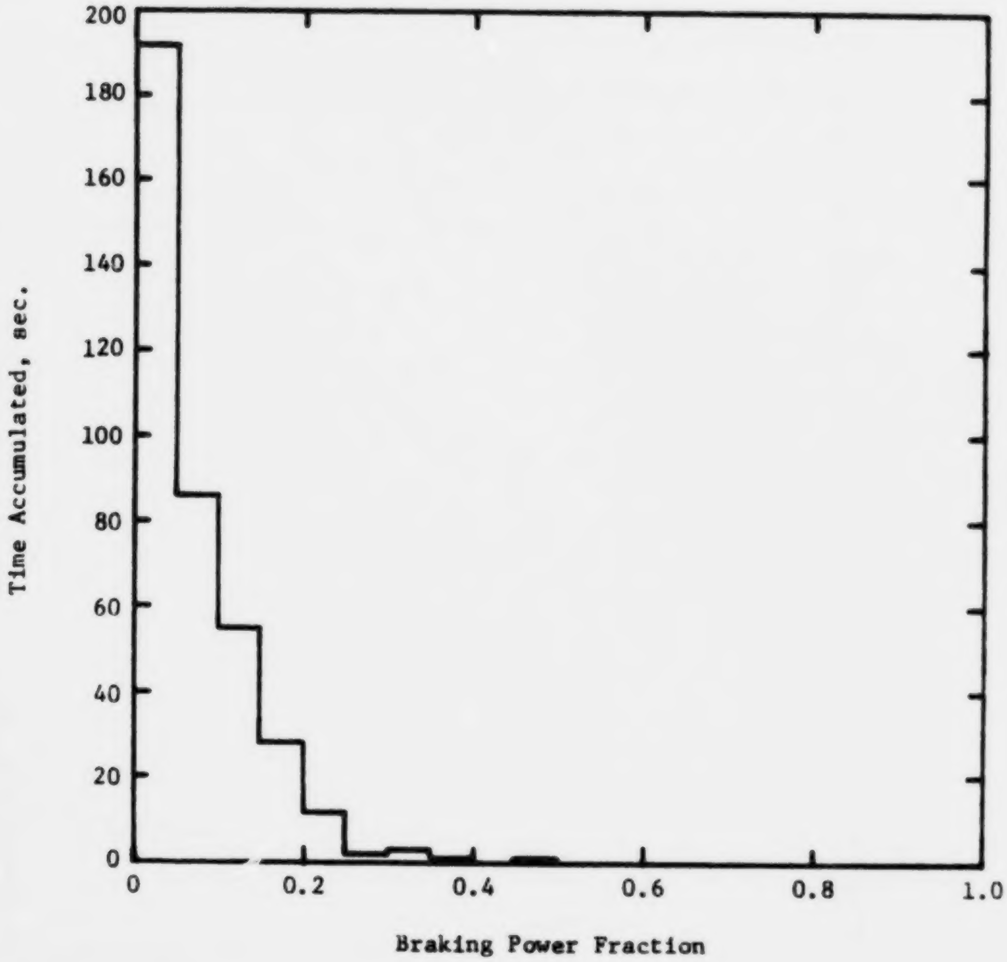


Figure 109. Braking Power Fraction Time Distribution on FDC for CD-2 Engine

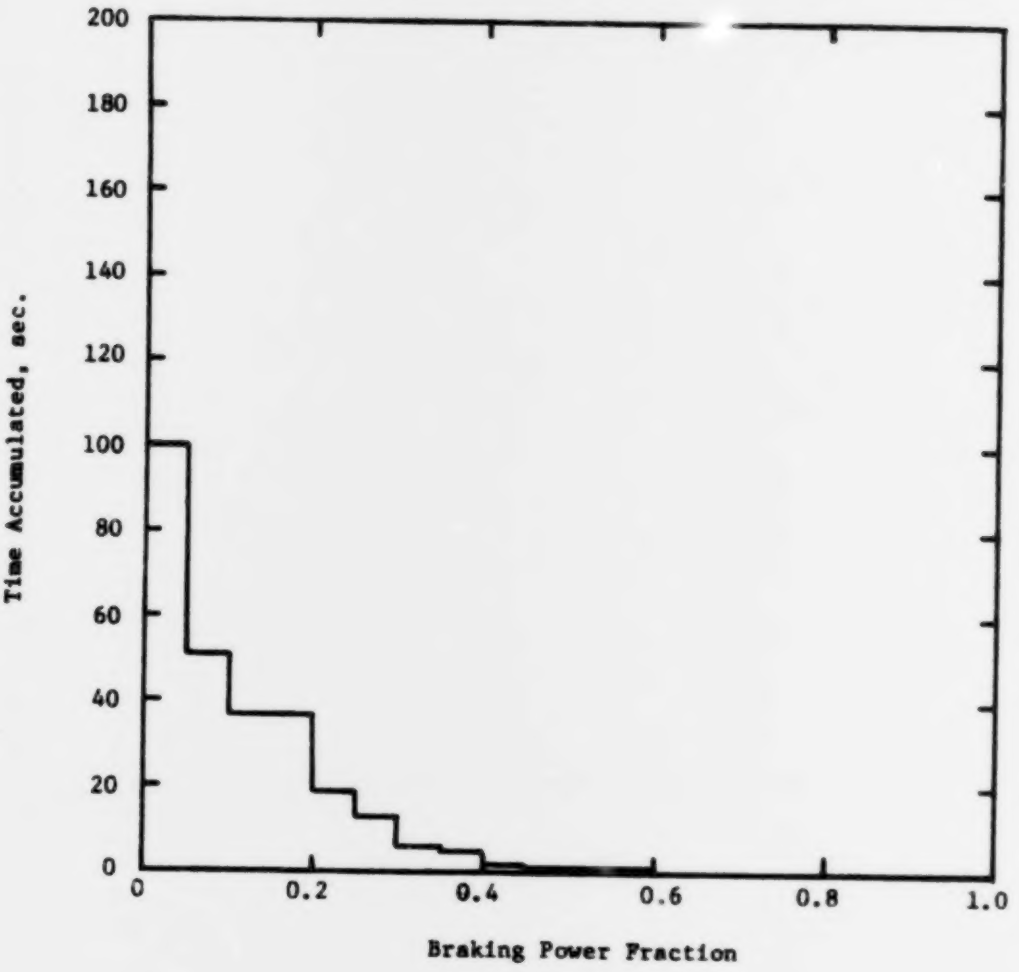


Figure 110. Braking Power Fraction Time Distribution for FDC on the PD-1 Engine

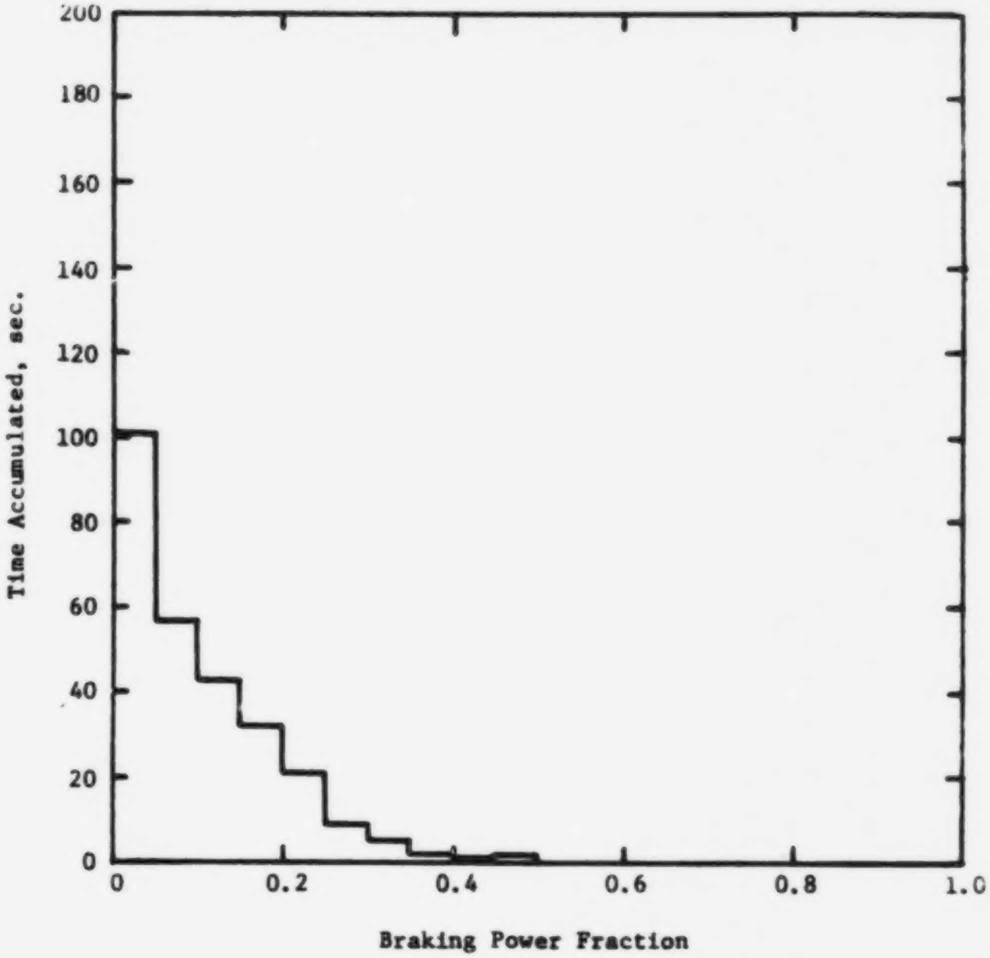


Figure 111. Braking Power Fraction Time Distribution on FDC for PD-2 Engine

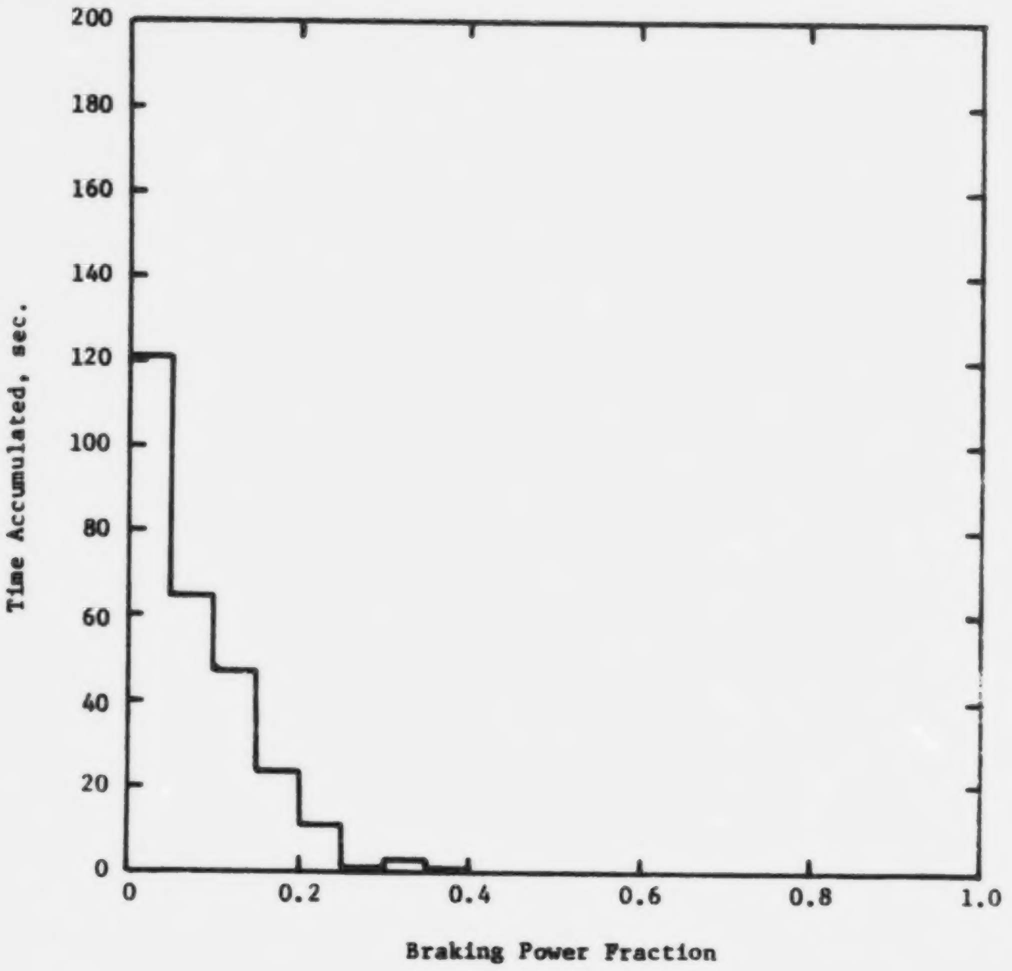


Figure 112. Braking Power Fraction Time Distribution on FDC for Internal Combustion Engine

engines but differ slightly due to inertia and speed range differences.

Shown in Figures 113, 114, 115, 116 and 117 are the transmission gear ratio histograms. For the CD-1 and PD-1 engines a Tracor infinitely variable transmission was used with a single-shaft engine and therefore, the histograms (Figures 113 and 115) are similar. Gear ratios around 2.0 accumulated the most time under power. At engine idle the engine shifts to the maximum gear ratio in anticipation of an acceleration. This accounts for the large time accumulation near the maximum gear ratio. The histogram for the CD-2 free-turbine engine (Figure 114) indicates a large time accumulation near the lower values of gear ratio, indicating the difference in operation between free-turbine and single-shaft engines. Since the ICE and PD-2 engine have conventional automatic transmissions the histograms are at discrete gear ratios.

The histograms of transmission input torque for the engines with Tracor transmissions are shown in Figures 118, 119 and 120. The histograms are similar and indicate the largest time accumulations in the low and mid torque ranges which is favorable to long life for these types of transmissions.

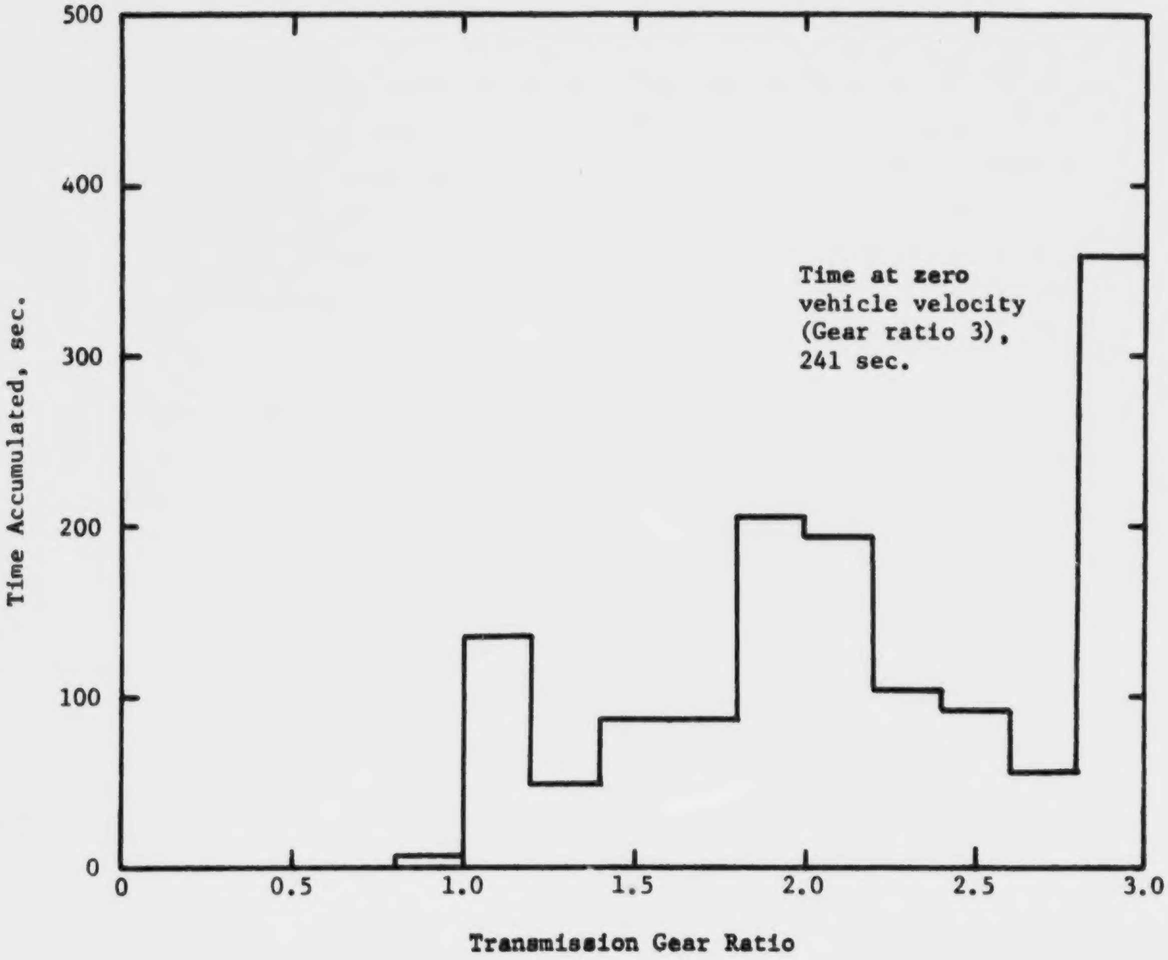


Figure 113. Transmission Gear Ratio Time Distribution on FDC with CD-1 Engine

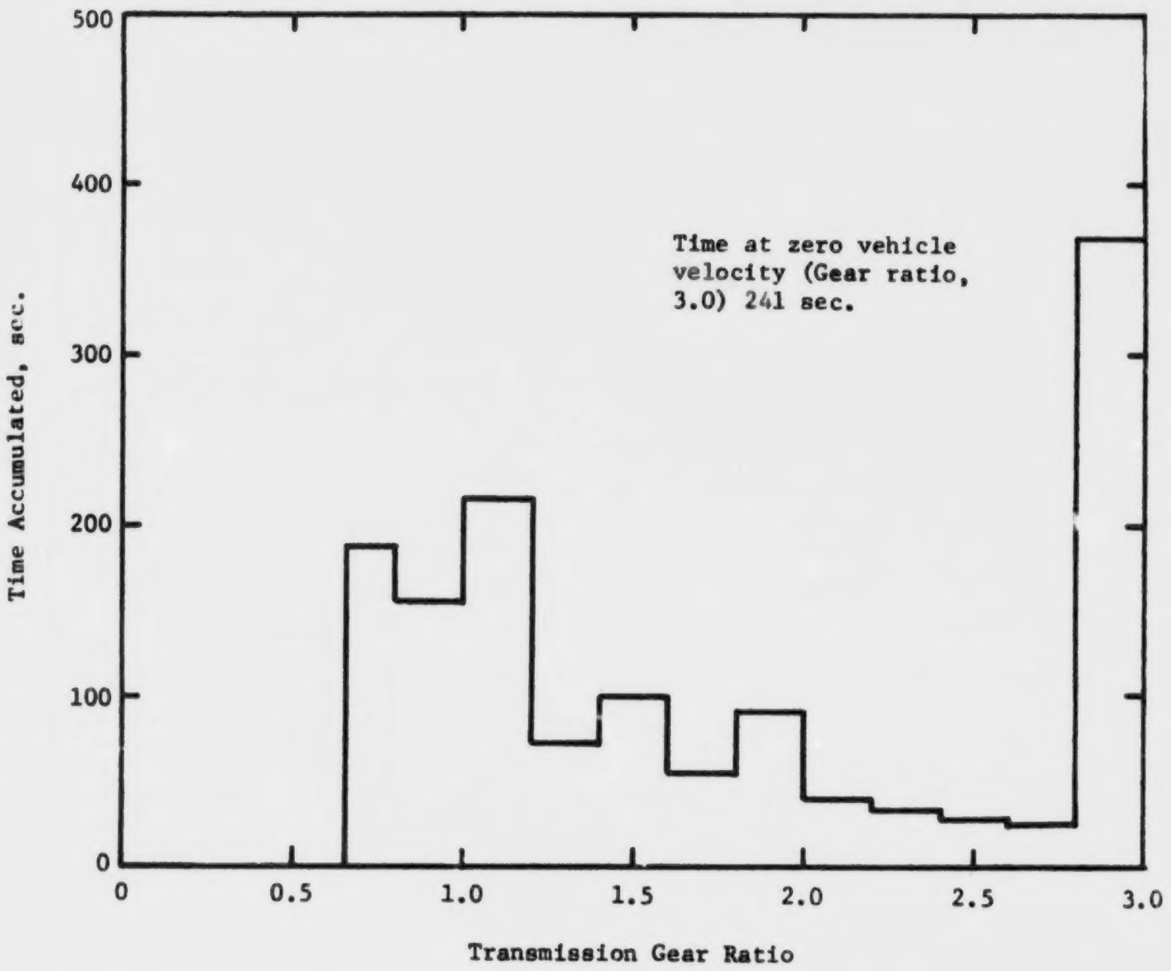


Figure 114. Transmission Gear Ratio Time Distribution on FDC with CD-2 Engine

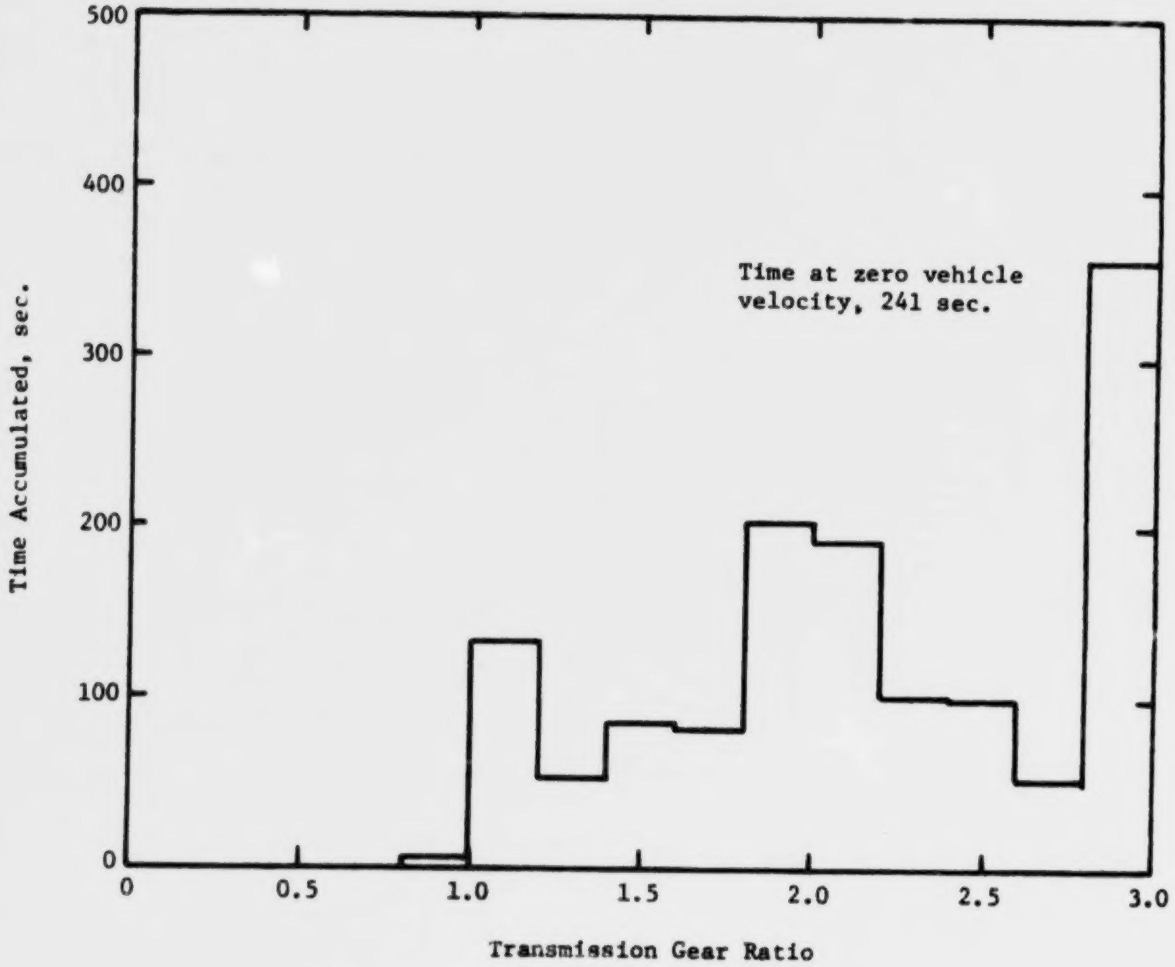


Figure 115. Transmission Gear Ratio Time Distribution on FDC with PD-1 Engine

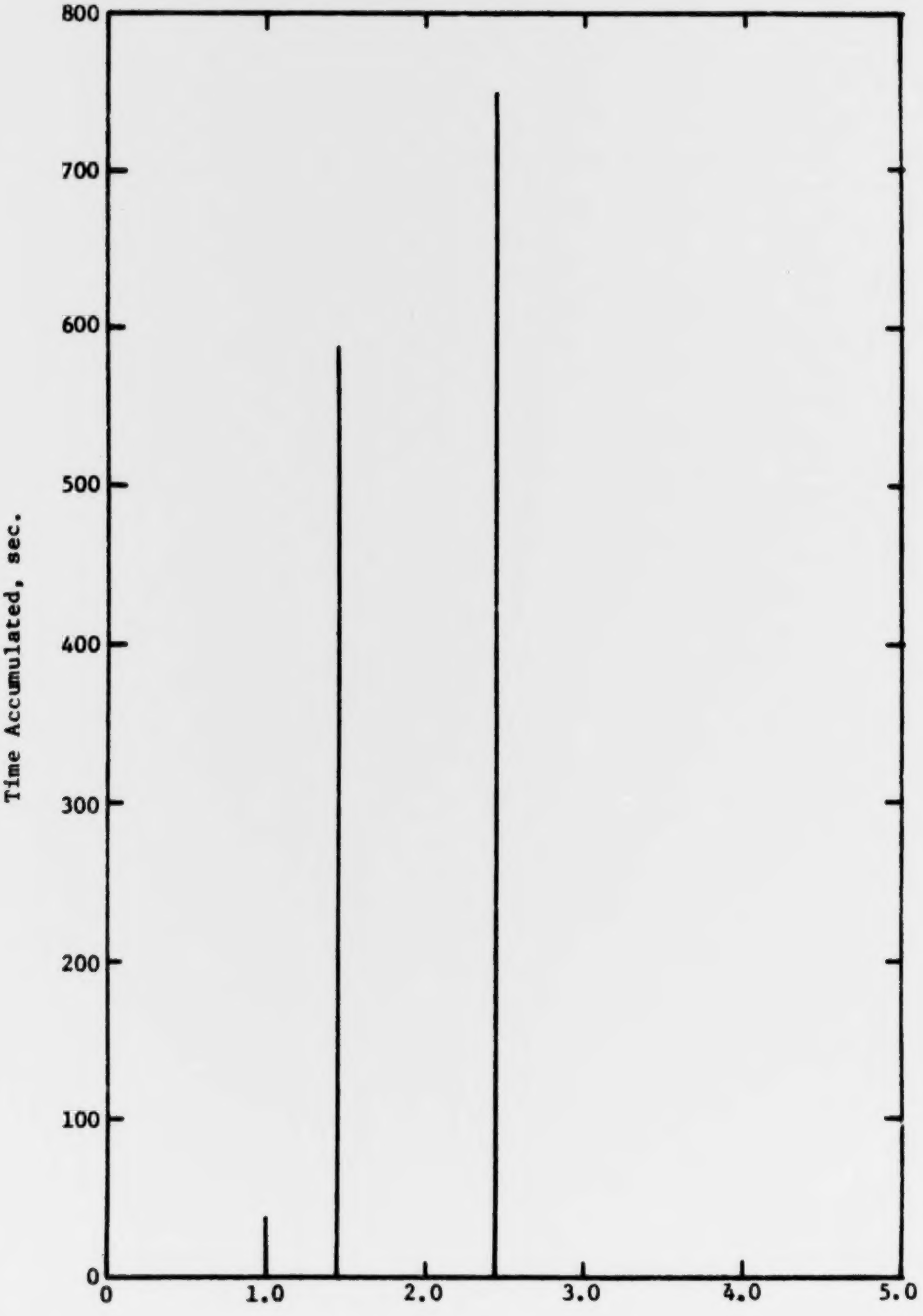


Figure 116. Transmission Gear Ratio Time Distribution on FDC for PD-2 Engine

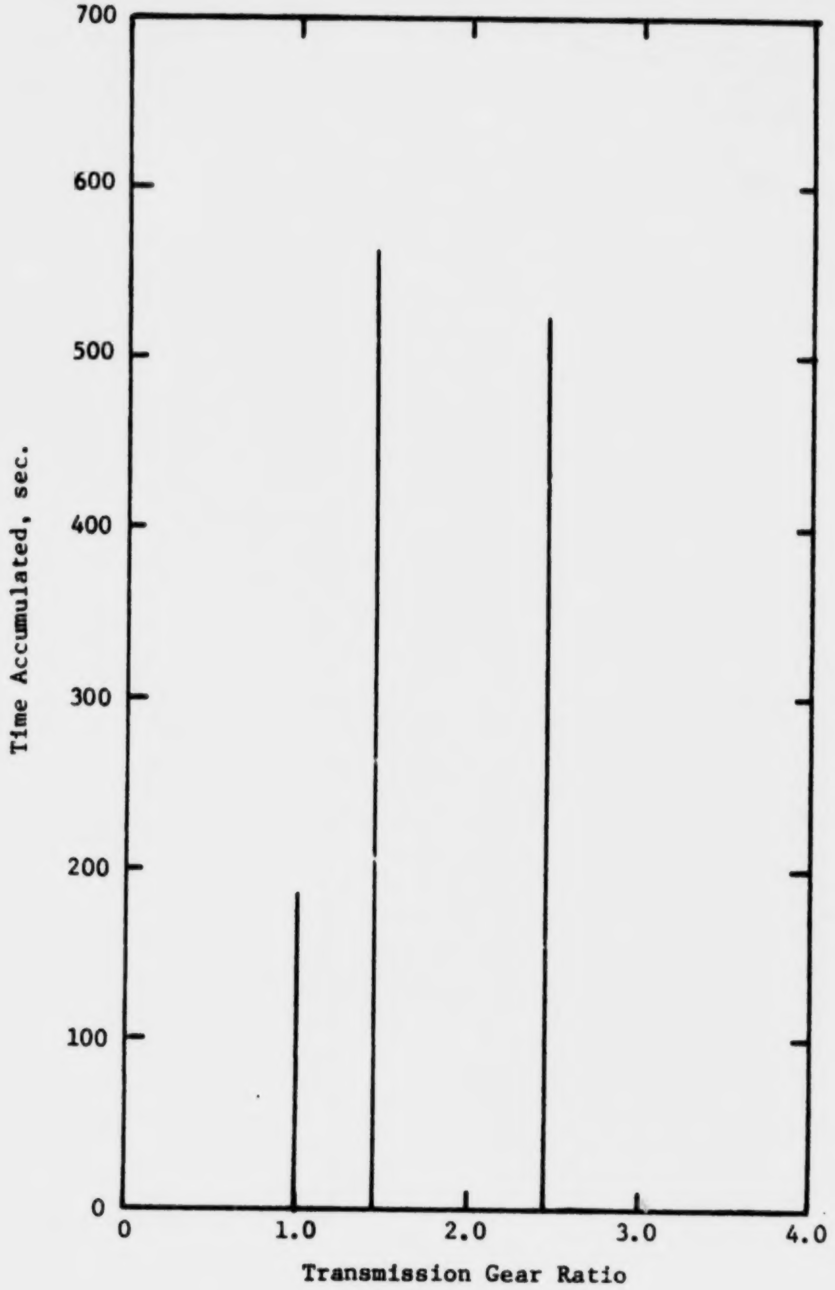
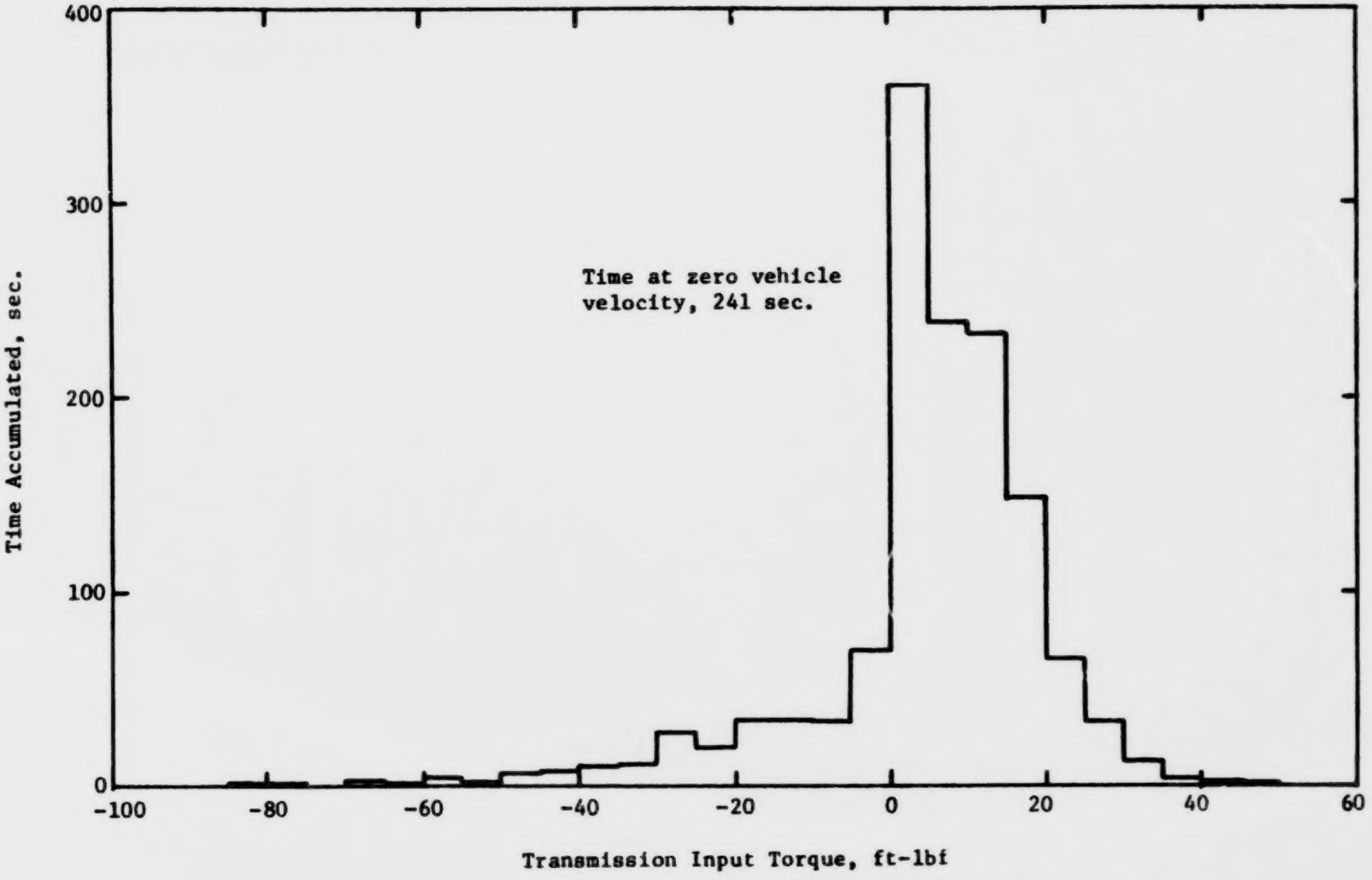


Figure 117. Transmission Gear Ratio-Time Distribution on FDC for Internal Combustion Engine



Time at zero vehicle velocity, 241 sec.

Figure 118. Transmission Input Torque Time Distribution on the FDC for the CD-1 Engine

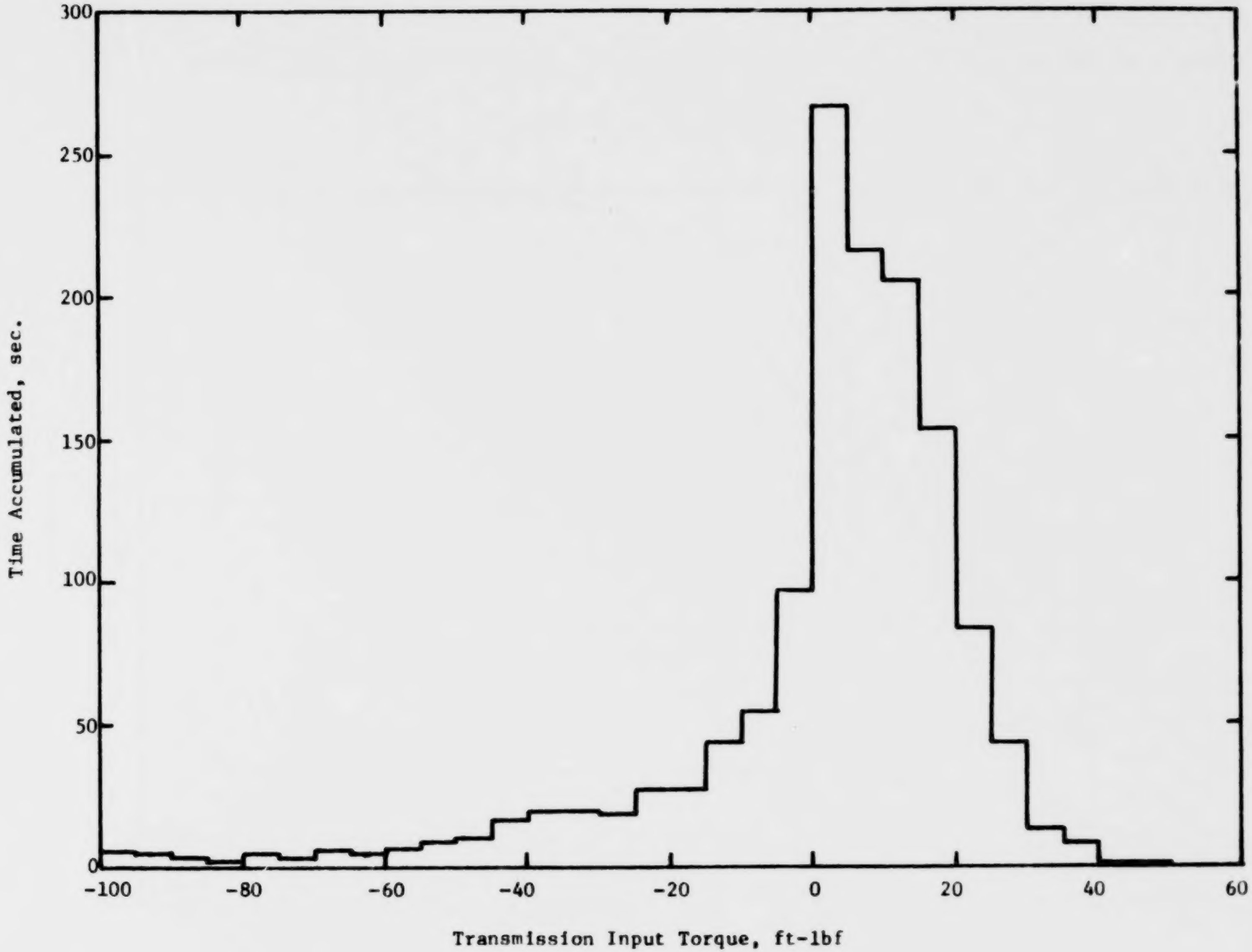


Figure 119. Transmission Input Torque Time Distribution on FDC for CD-2 Engine

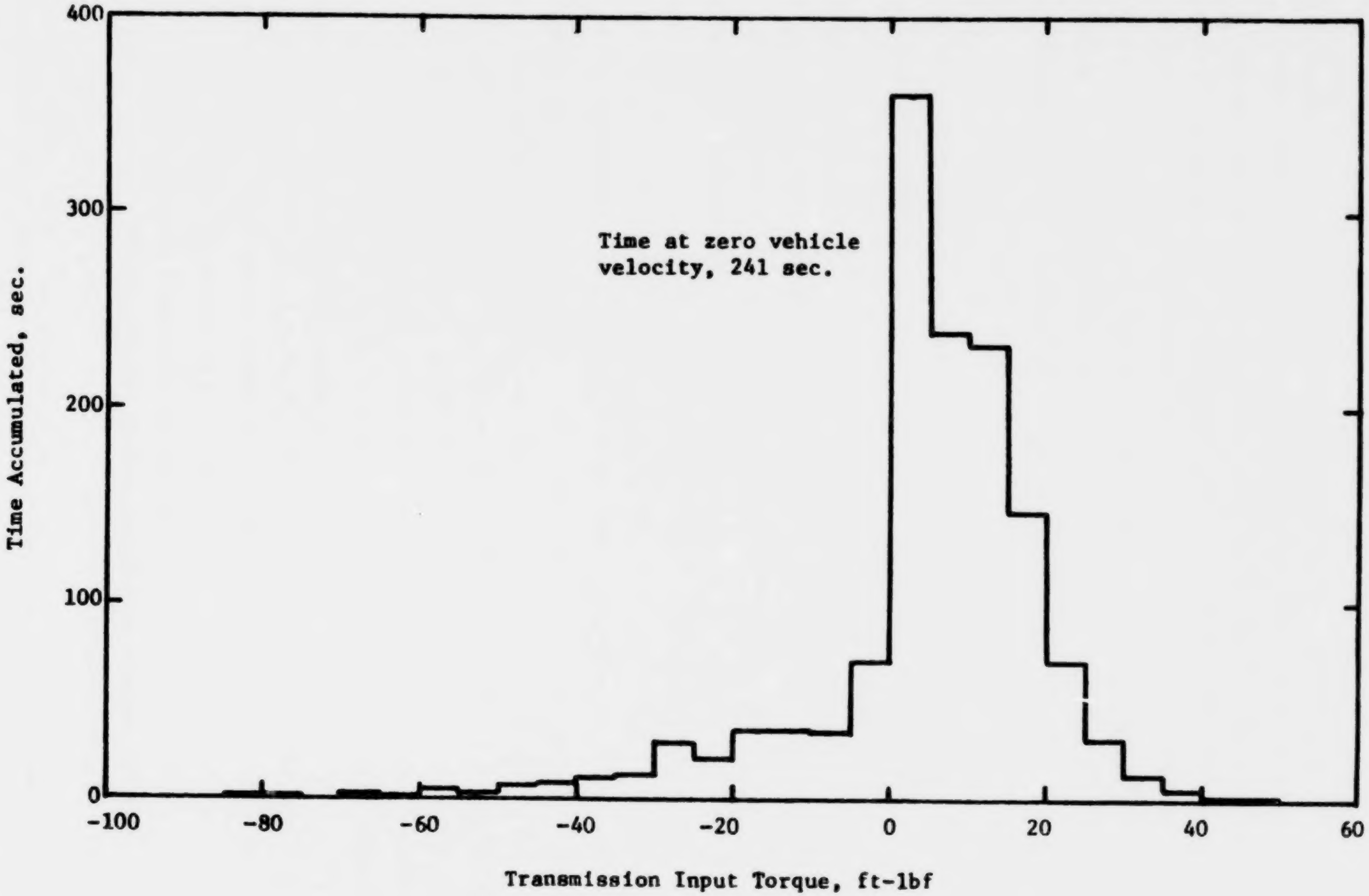


Figure 120. Transmission Input Torque Time Distribution on the FDC for the PD-1 Engine

5.7 Economic Analysis

This section of the program is concerned with the net-cost-of-ownership of automobiles with alternate versions of gas turbine engines as compared with conventional internal combustion engine powered vehicles. The basis of comparison is with a study (reference 22) made of the cost of operating an internal-combustion-engine-powered automobile for 100,000 miles in 10 years. The differential costs between the results of that study and those calculated for a gas turbine engine powered automobile are the net results of this economic analysis section.

As an introduction to this section, several points should be carefully noted, if the results of the study are not to be misinterpreted.

Shown in Table 22 are the cost of operating a 1970 standard automobile for 10 years. These costs are separated into those which can be modified because of engine selection, and these which are insensitive to engine type. In the present contract, there are several items which are specified at a different value than those in reference 22 specifically; the fuel cost per gallon, 31¢/gal vs.35¢/gal the total miles, 105,000 vs. 100,000; and, by implication, the gas mileage, which is computed in this study using the Oct. 15, 1971 Uniform Simplified Engine Duty Cycle, and which is 14.38 miles/gal vs. 13.8 miles/gal for the Internal Combustion Engine (ICE). In addition, the depreciation charge (cost less salvage) is split between that related to the engine and that related to the rest of the vehicle. The average cost per pound of the ICE powered vehicle is ∼ 85¢/lb, and a good estimate for the engine, transmission, starter, alternator, radiator, controls and exhaust system is about 900 pounds out of the 3800 lbs dry weight of the vehicle. If one assumes that the engine is slightly more expensive than the rest of the vehicle, say $1.00/lb, the retail cost of the above items is about $900 out of the $3185 shown in Table 22. This $900 represents an *estimate* of the *retail* cost of the specified portion of the propulsion system.

In the cost analysis of the gas turbine engines to follow, the *direct manufacturing* cost has been computed for each engine on the basis of 10^6 units per year. The ratio between the retail cost and the direct

Table 22

Estimated Cost of Operating an Automobile*
(1970 Internal Combustion Engine Powered)

Cost Element	Costs (10 years and 105,000 miles)		
Engine Related			
Engine initial cost (1)	$ 900		
Repairs and maintenance (2)	961		
Fuel (@ 31¢/gal)	2260		
Oil	160		
Total Engine related cost		$4281	
Vehicle Related			
Vehicle initial cost (1)	$2285		
Repairs and maintenance (2)	560		
Replacement tires	423		
Accessories	28		
Insurance	1722		
Garaging, parking, etc.	1805		
Taxes not included above	327		
Total Vehicle related costs		$7150	
Total Cost			$11,431

(1) Total Automobile cost = $3185

(2) Total Repairs and Maintenance = $1521

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*Derived from reference 22 with changes for this study.

manufacturing cost represents burden and profit all the way from the manufacturer to the ultimate customer. In addition, this factor influences the cost comparisons very strongly for the following reason. Given a calculated direct manufacturing cost for a turbine engine, a retail cost estimate for the comparable ICE engine, and the factor to convert direct manufacturing costs to retail costs, the ratio of retail costs of the gas turbine to the ICE is:

$$(\$AGT/\$ICE)_{retail} = \frac{(Factor) * (AGT\ Direct\ Cost)}{(ICE\ Retail\ Cost)}$$

This factor is highly proprietary to the automobile manufacturers for obvious reasons. Other sources, using a variety of methods, have arrived at factors between 1.90 and 3.45. A low factor tends to improve any comparisons while a high factor makes the gas turbine look poor. In essence, the unknown cost is the direct manufacturing cost of an ICE on the same basis as used in this study for the gas turbine engine. Any future comparisons between this study and others of a similar nature should investigate this matter rather thoroughly to be sure that a legitimate comparison is really being made.

The factor used for this study will be 2.0, which can be related (approximately) to the various markups as follows:

Dealer Markup (23%)	1.23
Manufacturer's Profit (13%)	1.13
Manufacturer's G/A Expense (20%)	1.20
Manufacturer's Overhead (20%)	1.20

Total Markup = 1.20 x 1.20 x 1.13 x 1.23 $\cong$ 2.0

The last item, overhead, is often figured as a percentage of the direct labor. When direct labor is very low, this percentage is quite large and can be over 400% to cover all labor not directly involved in making the engine.

It cannot be emphasized too strongly that this factor contains considerable uncertainty, and has direct effect on the final economic comparison. For this reason wherever possible cost comparisons among gas turbine engines should be made on the basis of direct labor and material.

3.7.1 Conceptual Design Engines

This section contains the results of the economic analysis of the two conceptual design engines, CD-1 (Single-Shaft) and CD-2 (Free-Turbine) (see Table 1). The analysis was done in less detail than for the succeeding two engines, and is primarily concerned with the direct cost of the engine and the fuel cost. Much of the accessory parts costs were taken from the latter portion of the study, since it was apparent that the costs of the two conceptual design engines could be substantially lowered.

3.7.1.1 Methods of Arriving at Costs

Power Plant Costs - In order to arrive at accurate and consistent cost values for the two selected cycles, the following procedure was followed. For each engine, assembly drawings (shown in the Appendix) were prepared along with parts lists. For each part a material selection was made, and the weight of the finished part calculated.

Using the drawings, parts lists, material selections and part weights for each engine, Manufacturing Engineering Services (MES), a component of the General Electric Company specializing in all aspects of mass production engineering, estimated the following items for each of the more than one hundred parts in each engine. First, the part weight, material, and original material form (casting, forging, plate, etc.) were used to calculate the material cost of each part in finished form. For parts to be purchased in a ready-to-use form (such as bearings and seals), the cost was determined from vendor quotes or prior cost experience. An allowance for scrap and waste was made depending on the form of the material. Second, for each part a manufacturing process was established which covered the operations to be carried out (drill, tap, spot face, etc.), the required machinery (multistation transfer machine, punch press, etc.), and the direct labor required to perform these operations. From this data, the direct labor required to fabricate each part was computed. Third, an assembly sequence for the entire engine was specified, and the required labor computed. From this data, the direct cost of the engine was calculated. In the process, the data on machinery requirements was generated so that capital costs could be estimated. (However, this was not done). Table 23 shows the cost calculation for one part

Table 23

Manufacturing Costs for Specimen Engine

Part Number 7, Compressor Front Shroud

Material Costs

Material	Nodular Iron Casting
Part Volume (Casting)	140 in^3
Part Weight	38 lbs.
Casting Cost/lb.	$0.45
Total Material Cost	$17.10

Manufacturing Cost (3 shift, 20 hours/day, 250 units/hour)

Operation	Equipment	Men/Shift
Chuck Turn OD & Shoulder Bore & Face 22" dia.	4 auto chuckers or Bullard	3
Face OD (2) Rear & Front 3" wide	3 Heald Borematics	3
Face-Small dia. groove Rear & Front		
Groove for O-rings		
Drill & Tap	Natco 2 station	1
Total Men		7
Cost @ $5.00/man hour		$0.168
TOTAL COST		$17.268

of one of the two conceptual design engines.

The result of this procedure is to arrive at the direct cost of the engine. Not included are the non-direct costs, and other charges which add up to the retail price, or cost of ownership charged against the engine proper. As mentioned previously, a factor of 2.0 times direct manufacturing cost is used to arrive at the retail cost allocated to the engine proper.

Each engine has a transmission selected to provide a good match between the engine and the vehicle. The costs for the infinitely variable transmissions were obtained from the company which has designed the particular units selected. Conventional transmissions were estimated from available data. No cost differential was assumed for the drive shaft and final drive between the various engines.

The cost of the controls for these engines is uncertain, since the exact requirements were not known at the time of the economic analysis. Rough estimates have been in the range of $100 not including the actuators which are costed for each engine. The control function operates between the driver demand, the vehicle speed, the engine parameters and the transmission. The choice between hydromechanical and electronic control cannot be made at this time, but if an electronic control using integrated circuits can be coupled with a simple hydraulic system, the cost should drop to the order of $50 in the production range specified.

The auxiliaries not costed for the engine proper,or the transmission, or the control system include the fuel pump, ignition system, alternator, and starter. Except for the ignition system, no cost difference exists between any of the engines studied (including the ICE). Costs for these items were estimated from available data at the direct cost level.

In all cases of powerplant component part cost, an effort has been made to distinguish between those parts which an automobile manufacturer would probably make himself and those he would purchase. At the present time, the automobile companies make castings, forgings, stampings, and machined parts. They purchase fasteners, bearings, starters and alternators.

Vehicle Cost - The vehicle cost for this study is the 1970 cost (Table 22) of $3185 less the assumed cost of the engine at the retail level of $900 or $2285. This cost will be unchanged throughout the study.

Fuel Cost - The fuel cost for each engine was calculated over the Uniform Simplified Engine Duty Cycle (Oct. 15, 1971) for a route of 105,000 miles. The results of these calculations are reported in Section 3.6 with the cost of fuel set at 31¢/gal. This is an arbitrary value, used only for consistency between various studies.

Salvage Value - The differential salvage value between an ICE and any gas turbine engine is a function of the amount of strategic or valuable material in the engine and the cost of reclaiming it. At present, the scrap value of entire automobiles is quite low. Reference 22 assumes no value as scrap, however, in large cities where central processing plants exist, a retail value of perhaps $50 may be assigned to the entire vehicle as scrap, with perhaps $12 being due to the engine. For each gas turbine engine, the additional scrap value of materials such as Inconel 713LC and Hastelloy X will be figured assuming some cost to separate the parts from the iron, steel and aluminum parts.

3.7.1.2 Results

As indicated above, the direct manufacturing costs for the two conceptual design engines were quite accurately calculated. However, the costs for auxiliary components were not, nor was a strong effort made to reduce the first-cut costs. The knowledge gained from this section was, however, used to reduce the costs of the two preliminary design engines.

3.7.1.2.1 CD-1 (Single Shaft) Engine

Engine Costs - In Table 24 is shown the elements of the costs for the CD-1 Single Shaft engine. As can be seen, this engine is more than twice the cost of an ICE ($900). Furthermore, the engine is relatively heavy. Shown in Table 25 are the weights for this engine. Table 26 shows a summary of the more expensive parts. As can be clearly seen,

Table 24

Single Shaft Engine Cost - CD-1

Labor Costs (Direct Labor)			
Manufacturing Costs	8.54		
Assembly Costs	4.89		
Total Labor Costs		13.43	
Material Costs			
Direct Material Costs	688.60		
Scrap, Chips, etc.	82.67		
Purchased Parts	32.60		
Total Material Costs		803.87	
Total Direct Manufacturers Cost			817.30
Accessory Costs			
Ignition System	2.00		
Starter & Alternator	20.00		
Fuel & Engine Control	50.00		
Transmission(TRACOR + Clutch + 2 Speed)	140.00		
Total Accessory Costs		212.00	
Manufacturer's Engine and Transmission Cost			1029.30
Markup to Retail			1029.30
Retail Engine and Transmission Cost			2058.60

Table 25

Single Shaft Engine Weights - CD-1

Engine Weight			
Basic Engine			
Manufactured Parts	580		
Purchased Parts	14		
Basic Weight		594	
Transmission		120	
Accessories			
Ignition System	3		
Starter & Alternator	30		
Fuel & Engine Control	5		
Accessory Weight		38	
Total Weight			752

Table 26

Single Shaft Engine Expensive Parts, CD-1

Parts	Material	Weight (lbs)	Cost/1b	Total Cost
Turbine Wheel	Inconel 738	16.5	5.50	90.75
Compressor Rotor	Aluminum	2.84	9.50	21.28
Turbine Scroll Shroud	Hastelloy X	23.98	3.00	72.00
All Other Parts	Inc. & Hast.	30.2	-	169.45
Main Housing	Nodular Iron	146.2	0.30	43.85
Regenerator	Cercor Ceramic	16.1	-	70.00
Expensive Parts Total		235.7		467.33
% of Total Engine		39.7		58.2

the large turbine and compressor, together with the high cost of these precision castings,has a significant effect on total cost. One of the results of this analysis was to influence the design of the subsequent engines; for example, the turbine wheel in the PD-1A engine,also single shaft, weighs only 7.5 pounds versus the 16.5 pounds in the CD-1 engine, a saving of $50 at the manufacturing level, and $100 at the retail level.

A discussion of further cost reductions will be postponed until a later section.

Vehicle Cost - As noted earlier, the vehicle cost is assumed to be $2285 for this study.

Fuel Cost - Based on the results over the specified driving cycle, the CD-1 engine has an average fuel mileage of 16.06 miles/gal for a total cost of $2025 for 105,000 miles.

Salvage Value - The CD-1 engine contains 71 pounds of Inconel 738 and Hastelloy X. Various contacts have established that these alloys have a potential value as scrap, in large quantities, of as much as 60¢/lb. This would imply that about $42 worth of high-nickel alloy scrap in addition to the other materials such as stainless steel, is available.

With the machinery available at a large processing plant, it should be possible to strip the larger parts and recover them at a cost of only a few dollars an engine. Assuming that the stainless steel (63 lbs) is worth a few dollars, the additional salvage value of this engine might be about $40.

It is specifically not proposed to reuse any parts. A vehicle scrapped before the end of 10 years life might be treated in this way, the common practice at present, but no such assumption is made here.

Net Cost of Ownership - For this engine, no separate calculation of engine repairs, maintenance, or oil consumption were made. The values determined in the Preliminary Design section were therefore used. The net cost of ownership will be shown in a later section where these values are discussed. The original cost is $4344 versus $3185 for the 1970 ICE, an increase of 36.4%. The lower maintenance costs and better gas mileage cannot be expected to offset this increase, although the cost of reducing emissions of a 1970 ICE to that required by 1975 or 1976 have been estimated (reference 23) at $240 - $500. Thus, the CD-1 engine appears significantly more expensive with respect to first cost.

3.7.1.2.2 CD-2 (Free Turbine)

Engine Cost - Table 27 shows the cost elements for the CD-2 Free Turbine engine. This engine, like the CD-1, is more expensive than the ICE. As shown in Table 28, it is heavier than the CD-1 engine, but because of the smaller amount of Inconel and Hastelloy X, the net cost is somewhat lower. The expensive parts are shown in Table 29.

Vehicle Cost - As noted earlier, the vehicle cost is assumed to be $2285.

Fuel Cost - The average mileage over the specified driving cycle was 15.39 miles/gal for a total cost of $2114 for 105,000 miles.

Salvage Value - The CD-2 engine contains 44 pounds of high nickel alloy, much less than the 71 pounds in the CD-1 engine. Assuming the same 60¢/lb value as separated scrap, the additional scrap value could be as high as $26. In addition, the CD-2 has 84 pounds of stainless steel.

Net Cost of Ownership - This engine is cheaper than the CD-1, but still more expensive than the ICE. The original vehicle cost is $4074 versus $3185 for the 1970 ICE engine and $4344 for the CD-1 powered vehicle. The same comments as were made for the CD-1 apply to the CD-2, namely that the CD-2 engine will be significantly more expensive with respect to original cost than the ICE.

Table 27

Free Turbine Engine Cost - CD-2

Labor Costs (Direct Labor)			
Manufacturing Costs	8.47		
Assembly Costs	4.67		
Total Labor Cost		13.19	
Material Costs			
Direct Material Costs	591.09		
Scrap, Chips, etc.	66.13		
Purchased Parts	68.34		
Total Material Cost		725.56	
Total Direct Manufacturer's Cost			738.70
Accessory Cost			
Ignition System	2.00		
Starter & Alternator	20.00		
Fuel & Engine Control	50.00		
Transmission (TRACOR)	84.00		
Total Accessory Cost		156.00	
Manufacturer's Engine and Transmission Cost			894.70
Markup to Retail			894.70
Retail Engine and Transmission Cost			1789.40

Table 28

Free Turbine Engine Weights - CD-2

Engine Weights			
Basic Engine			
Manufactured Parts	635		
Purchased Parts	20		
Basic Weight		655	
Transmission		73	
Accessories			
Ignition System	3		
Starter & Alternator	30		
Fuel & Engine Control	5		
		38	
Total Engine and Transmission Weight			766

Table 29

Free-Turbine Engine Expensive Parts, CD-2

Parts	Material	Weight (lbs)	Cost/lb	Total Cost
Gas Generator Turbine	Inconel 738	4.80	5.50	26.46
Free Turbine	CRM 6D	8.36	2.77	23.16
Compressor Rotor	Aluminum	1.27	9.50	12.06
Turbine Diffuser Shroud	Hastelloy X	12.22	4.55	55.50
Gas Generator Turbine Scroll	MAR M 507	6.61	4.00	26.46
All Other Parts	Inc. & Hast.	11.59	-	66.40
Main Housing	Nodular Iron	201.10	0.30	60.32
Regenerator	Cercor Ceramic	16.10	-	70.00
Expensive Parts Total		262.05		340.36
% of Engine Total		40.0		46.9

3.7.2 Preliminary Design Engines

Since the results of the basic engine cost analysis were available during the preliminary design effort, a significant amount of insight into these areas where cost improvement could be made was applied. Three areas were particularly important.

First, the rotary regenerators at $35 each were a significant cost item for both conceptual design engines. On a per pound basis, this is approximately $4.50/lb, or more than the specific cost of all but the highest cost alloys. Revised estimates for these parts, at $30/unit, as finished pieces,were obtained for the PD-1 engine.

The recuperator in the PD-2 engine was quoted at $35/cubic foot as an unfinished part. Discussions with one vendor who makes both types of units (rotary regenerators and cross flow recuperators) indicated that the finished cost of both types of units could be nearly the same. Because the cost of both types of unit is a large fraction of the total cost, and in order not to bias the comparison between these two engines, the cost of both units will be set at $30/unit for this study.

Second, the high cost of the Inconel and Hastelloy parts is related to the component size as well as to the material cost. A great deal of effort was made to reduce the size of these parts, which was moderately successful. The specific costs, however, were not changed as they seemed reasonable.

Third, the use of massive nodular iron castings,which was characteristic of both the CD-1 and CD-2 engines, was investigated. In many cases, aluminum die castings could be sutstituted, lowering both the weight and cost. The cost reduction was due partially to the lower density of aluminum, and partially to the smaller wall thickness permitted by the die casting process.

3.7.2.1 Methods of Arriving at Costs

Engine Cost - The engine costs for both preliminary design engines were generated in a similar fashion to that used for the conceptual design engines. The only difference was in the treatment of the labor cost.

It was observed that direct labor was a very small portion of the cost of both the CD-1 and CD-2 engines, about $14 out of $1000. It did not appear worthwhile to recalculate labor costs for the two preliminary design engines since even a large variation in labor cost would not be a significant variation in the total cost. Hence, an estimate of $13.50 was used for the labor charge for both the PD-1 and PD-2 engine.

Certain parts, such as the regenerator (PD-1) and recuperator (PD-2) were considered purchased parts, thus changing the split between the so-called direct manufacturing cost and the purchased parts cost.

A digital computer was used to prepare the parts lists for these engines. These lists appear in Appendix A. This procedure aided in keeping track of the engine details and facilitated making changes.

Vehicle Costs, Fuel Costs, and Salvage Value - No changes in procedure were made for these engines.

Maintenance and Repair - The 1970 report on the Cost of Operating an Automobile lists $1521 as the ten-year cost of repairs and maintenance. It includes lubrication; washing and waxing; replacement of spark plugs, points and condenser, wiper blades, fan belt(s), radiator hoses; starter, water pump, and brake overhaul; universal joint replacement; and major repairs such as a "valve job". The procedure used in this study was to break the $1521 into two portions, one associated with the vehicle which was essentially independent of engine type, and the other engine type dependent. Some adjustment of the actual values and the assignment between vehicle and engine were required to force the total to equal $1521. Table 30 shows these results for a 1970 ICE powered automobile. As expected, except for accidents, most repairs and maintenance is engine related, and will become more so as the emission control devices become more elaborate.

The gas turbine engine will require significantly lower maintenance than the ICE. Table 31 shows the assumed maintenance and repairs used in this study. The "12,000 mile" service is primarily to clean the air filter and generally check the engine superficially. The "24,000 mile" service covers the replacement of the air and oil filter, a check on fuel

Table 30

Maintenance and Repair - Internal Combustion Engine

Item	Interval	Cost	Vehicle or Engine Cost	Total Cost (10 years & 105,000 miles)
Maintenance				
"6000 mile" Service	6000 miles or 6 mos.	$ 5	Engine	$ 100
"12000 mile" Service	12000 miles or 12 mos.	40	Engine	400
"24000 mile" Service	24000 miles or 2 years	15	Engine	75
"36000 mile" Service	36000 miles or 3 years	30	Vehicle	90
Wash and Wax (average over 10 yrs)	6 months	2	Vehicle	40
Total Maintenance				705
Repairs				
Major Repair ("Valve Job")	Once near end of life	246	Engine	246
Reline Brakes	30,000 miles	65	Vehicle	195
Replace Water Pump	50,000 miles	30	Engine	60
Replace Battery	3 years	35	Vehicle	105
Repair Starter and/or alternator	50,000 miles	20	Vehicle	40
Replace Fan Belts	30,000 miles	10	Vehicle	30
Replace Universal Joints	50,000 miles	30	Vehicle	60
Replace Muffler	50,000 miles	40	Engine	80
Total Repairs				816

Total Repairs & Maintenance = $1521

Total Engine Related = 961

Total Vehicle Related = 560

Table 31

Maintenance and Repair - Gas Turbine Engines

	Engine Only		Total Cost (10 years & 105,000 miles)	
Item	Interval	Cost		
Maintenance				
"12000 mile" Service	12000 miles or 1 year	10	100	
"24000 mile" Service	24000 miles or 2 years	40	200	
Total Engine Maintenance			300	
Repairs				
Repair Hydraulic Actuator System	50,000	60	120	
Replace Regenerator Seals (Not PD-2)	50,000	50	100	
Replace Silencer	One Time near end of life	50	50	
Repair Combustor	50,000	50	100	
Total Engine Repair			370	
Total Engine Repairs and Maintenance				670

control operation, and possibly a compressor cleaning operation.

No major repair is included in this tabulation as the failure patterns of gas turbines is quite different from that of the internal combustion engine. The repairs include a rebuilding of the hydraulic system, similar to a brake job, the replacement of the regenerator seals on the CD-1, CD-2 and PD-1 engines, a silencer replacement (this may not be necessary), and a combustor repair or replacement. The final costs are significantly lower than those of an internal combustion engine.

Oil cost was estimated for the turbine engine on the basis of 5 quart capacity, 24,000 miles or two year change, with a filter change (charged to maintenance) at 24,000 miles. The cost of the oil was assumed to be higher than presently paid, i.e., $1.50/quart. This oil cost is thus $37.50, rounded to $40 for tabulation.

3.7.2.2 Results

3.7.2.2.1 PD-1 Single Shaft

Engine Cost - The results of the cost study on the PD-1 engine are shown in Table 32, and the corresponding engine weights are shown in Table 33. Note that this engine is lighter than either of the previous designs as well as less expensive. Table 34 is a computer listing of all parts with a total cost greater than $5.00. Note that out of a total of $503 (basic cost less scrap), these parts account for $424.

Appendix A contains the drawings used for the cost study as well as complete detailed parts lists. These lists include the materials, specific costs, and total costs. Separate lists show the breakdown by material and form.

Vehicle Cost - The vehicle cost is the same as for all other cases, $2285.

Fuel Cost - The fuel cost for the PD-1 engine is $2391 for 105,000 miles, at a fuel mileage of 13.59 miles/gal.

Table 32

Single Shaft Engine Costs - PD-1

Labor Costs (Direct Labor)		13.50	
Material Costs			
Direct Material Costs	349.61		
Scrap, Chips, etc.	40.96		
Purchased Parts	153.61		
Total Material Costs		544.18	
Total Direct Manufacturer's Cost			557.68
Accessory Costs			
Ignition System	2.00		
Starter & Alternator	20.00		
Fuel and Engine Control	50.00		
Transmission(TRACOR + Clutch + 2 Speed)	140.00		
Total Accessory Cost		212.00	
Manufacturer's Engine and Transmission Cost			769.68
Markup to Retail			769.68
Retail Engine and Transmission Cost			1539.36

Table 33

Single Shaft Engine Weights - PD-1

Engine Weights			
Basic Engine			
Manufactured Parts	413		
Purchased Parts	71		
Basic Weight		484	
Transmission		120	
Accessories			
Ignition System	3		
Starter & Alternator	30		
Fuel and Engine Control	5		
Accessory Weight		38	
Total Engine Weight			642

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Table 34

Parts With Cost Greater Than $5.00

AUTOMOTIVE GAS TURBINE SELECTION STUDY
PARTS LIST
PD-1 ENGINE (SINGLE SHAFT)

PART	NAME	DESCRIPTION	MATERIAL	TYPE	COST/LB	WEIGHT	TOTAL WEIGHT	NUMBER OF PARTS	COST PER PART	TOTAL COST
003	REGENERATOR	-	CERCOR	PUR	0.00	10.000	20.00	2	30.00	60.00
030	VANES	TURBINE STATOR	INC 713LC	PIC	10.00	0.051	5.10	100	0.51	51.00
001	TURBINE	-	INC 713LC	PIC	5.50	7.500	7.50	1	41.25	41.25
041	SUP FRAME	CENTER	NOD. IRON	CST	0.20	140.000	140.00	1	28.00	28.00
026B	SHELL	TURB INLET EXHAUST OUTER	HAST. X	PIC	6.00	4.500	4.50	1	27.00	27.00
002	COMPRESSOR	-	C355-T61 AL	PIC	9.50	2.500	2.50	1	23.75	23.75
038	FRIT	COMBUSTOR AND SUPPORTS	-	PUR	0.00	7.000	7.00	1	19.00	19.00
053	TIP SEAL	TURBINE	HAST. X	HNC	134.00	0.126	0.13	1	16.88	16.88
026A	SHELL	TURB INLET EXHAUST OUTER	304 SST	PIC	2.00	8.250	8.25	1	16.50	16.50
027	LINK	NOZZLE VANE ACTUATOR	304 SST	PIC	2.00	0.084	6.05	72	0.17	12.10
049	STRUT	-	HAST. X	BAR	3.00	0.475	3.80	8	1.42	11.40
112	BEARING	REG. DRIVE SPROCKET	-	PUR	0.00	0.122	0.49	4	2.80	11.20
005	COVER PLATE	REGENERATOR	NOD. IRON	CST	0.25	17.000	34.00	2	4.25	8.50
068	BEARING	TRANSMISSION DRIVE GEAR	-	PUR	0.00	0.745	1.49	2	3.80	7.60
092	SEAL PLATES	REGENERATOR	304 SS	EXT	0.70	5.123	10.25	2	3.59	7.17
078	SCROLL	COMPRESSOR INNER	43 AL	DCS	0.45	15.000	15.00	1	6.75	6.75
011	BEARING	MAIN SHAFT	-	PUR	0.00	0.522	1.04	2	3.25	6.50
023	HOUSING	MAIN BEARING	NOD. IRON	CST	0.25	26.000	26.00	1	6.50	6.50
094	PLATE	REGENERATOR	304 SS	EXT	0.70	4.600	9.20	2	3.22	6.44
054	ACT RING	TURBINE NOZZLE	304 SST	CST	2.00	3.100	3.10	1	6.20	6.20
007	FLANGE	TURBINE INLET REGENERATOR	304 SST	CST	2.00	3.000	3.00	1	6.00	6.00
042	SCROLL	COMPRESSOR OUTER	333 AL	DCS	0.55	10.600	10.60	1	5.83	5.83
012	PLATE	COMB FWD END-SUP PLENUM	HAST. X	SHT	3.00	1.900	1.90	1	5.70	5.70
024	BEARING	MAIN DRIVE GEAR	-	PUR	0.00	0.339	0.68	2	2.80	5.60
046	BEARING	AUX. SHAFT DRIVE GEAR	-	PUR	0.00	0.339	0.68	2	2.80	5.60
066	BEARING	AUXILIARY SHAFT	-	PUR	0.00	1.444	2.89	2	2.80	5.60
071	BEARING	CHAIN BELT SHAFT	-	PUR	0.00	0.544	1.09	2	2.80	5.60
089	CHAIN	REG.DRIVE GEAR	-	PUR	0.00	1.019	2.04	2	2.50	5.00
103	CHAINBELT	SPEED REDUCER	-	PUR	0.00	0.664	1.33	2	2.50	5.00

TOTAL WEIGHT = 329.59 LBS.   TOTAL COST = $423.67   NUMBER OF PARTS = 221

Salvage Value - There are 23 pounds of Inconel and Hastelloy in the PD-1 engine. Using the same logic as before, the incremental salvage value is $14 over the internal combustion engine.

Repairs, Maintenance and Oil Cost - These items were discussed previously. For the PD-1 engine, the repair and maintenance cost is $670 and the oil cost is $40.

3.7.2.2.2 PD-2 Free-Turbine

Engine Cost - The results of the cost study on the PD-2 engine are shown in Table 35, and the corresponding engine weights are shown in Table 36. Note that this engine is cheaper than the PD-1. Table 37 shows all parts with a cost greater than $5.00. Out of a total basic cost (less scrap) of $432, these expensive parts account for $354.

The complete parts and materials lists, along with the ballooned drawings are located in Appendix A.

Vehicle Cost - The vehicle cost is the same as for all three cases, $2285.

Fuel Cost - The fuel cost for the PD-2 engine is $2635 for 105,000 miles, at a fuel mileage of 12.34 miles/gal.

Salvage Value - There are only 11 pounds of Inconel and Hastelloy in the PD-2 engine, one reason its cost is as low as it is. The salvage value of this material, using the previously described logic, is only $7 greater than that of the internal combustion engine.

Repairs, Maintenance and Oil Cost - As previously discussed, these costs are $570 for repairs and maintenance, down $100 from PD-1 because of the lack of the need to replace the regenerator seals. The oil costs is the same as previously estimated, $40.

3.7.3 Summary

The results of this section of the study are shown in Table 38. With respect to net cost of ownership, the variation between turbine engine types is not great and all are comfortably under 110% of the 1970

Table 35

Free Turbine Engine Costs - PD-2

Labor Costs (Direct Labor)		13.50	
Material Costs			
Direct Material Costs	309.68		
Scrap, Chips, etc.	40.19		
Purchased Parts	122.64		
Total Material Costs		472.51	
Total Direct Manufacturer's Cost			486.01
Accessory Costs			
Ignition System	2.00		
Starter and Alternator	20.00		
Fuel and Engine Control	50.00		
Transmission	172.00		
Total Accessory Cost		244.00	
Manufacturer's Engine and Transmission Cost			730.01
Markup to Retail			730.01
Retail Engine and Transmission Cost			1460.02

Table 36

Free Turbine Engine Weights - PD-2

Engine Weights			
Basic Engine			
Manufactured Parts	342		
Purchased Parts	83		
Basic Weight		425	
Transmission		150	
Accessories			
Ignition System	3		
Starter and Alternator	30		
Fuel and Engine Control	5		
		38	
Total Engine and Transmission Weight			613

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Table 37

Parts With Costs Greater Than $5.00

AUTOMOTIVE GAS TURBINE SELECTION STUDY
PARTS LIST
PD-2 ENGINE (FREE TURBINE)

PART	NAME	DESCRIPTION	MATERIAL	TYPE	COST/LB	WEIGHT	TOTAL WEIGHT	NUMBER OF PARTS	COST PER PART	TOTAL COST
059C	RECUPERATOR	(CORE)	CERVIT	PUR	0.00	21.728	43.46	2	30.00	60.00
080A	NOZ. HOUSING	GAS GEN TURBINE	HAST. X	CST	10.00	6.832	6.83	1	68.32	68.32
108	SCROLL	EXHAUST	304 SST	CST	2.00	21.702	21.70	1	43.40	43.40
059A	RECUPERATOR	(SHELL)	304 SST	SHT	0.70	26.172	52.34	2	18.32	36.64
103	CASING	POWER TURB OUTER	304 SST	CST	2.00	11.481	11.48	1	22.96	22.96
009	TURBINE	GAS GEN	INC 713LC	PIC	6.50	2.230	2.23	1	14.50	14.50
056	OUTER SHELL	COMBUSTOR	304 SST	CST	2.00	7.213	7.21	1	14.43	14.43
012	BEARING	STARTER GAS GEN GEAR SHAFT	-	PUR	0.00	0.122	0.49	4	2.80	11.20
025	BEARING	POWER TURBINE SHAFT	-	PUR	0.00	0.280	1.12	4	2.80	11.20
054	HOUSING	BEARING GAS GEN	NOD. IRON	CST	0.25	41.384	41.38	1	10.35	10.35
098	COVER	GEAR BOX	NOD. IRON	CST	0.25	36.632	36.63	1	9.16	9.16
067	TIP SEAL	GAS GEN TURBINE	HAST. X	HNC	70.80	0.122	0.12	1	8.64	8.64
087	HOUSING	BEARING POWER TURBINE	NOD. IRON	CST	0.25	28.776	28.78	1	7.19	7.19
005	COMPRESSOR	-	410 SST	CST	2.00	3.467	3.47	1	6.93	6.93
082	CASING	OUTER	B50T13	CST	0.30	22.350	22.35	1	6.70	6.70
068	CASING	TURBINE INNER	304 SST	CST	2.00	2.891	2.89	1	5.78	5.78
022	TURBINE	POWER	CMR-60	PIC	2.77	2.061	2.06	1	5.71	5.71
035	BEARING	ACCESSORY DRIVE GEAR	-	PUR	0.00	0.339	0.68	2	2.80	5.60
094	BEARING	OIL PUMP	-	PUR	0.00	0.339	0.68	2	2.80	5.60

TOTAL WEIGHT = 285.91 LBS. TOTAL COST = $354.31 NUMBER OF PARTS = 29

Table 38

Comparison Summary
(10 Years and 105,000)

Item	Cost in Dollars				
Engine Type	1970 ICE	CD-1	CD-2	PD-1A	PD-2A
			NET COST OF OWNERSHIP		
Engine Related Costs					
Engine Cost	$ 900	$ 2059	$ 1789	$ 1539*	$ 1460*
Repairs & Maintenance	961	670	670	670	570
Fuel	2260	2025	2114	2391	2635
Oil	160	40	40	40	40
Sub-Total	4281	4794	4613	4640	4705
less differential salvage	0	42	26	14	7
Total	4281	4752	4587	4626	4698
Vehicle Related Costs					
Vehicle Cost	2285				
Repairs & Maintenance	560				
Replacement Tires	423				
Accessories	28				
Insurance	1722				
Garaging, Parking, etc.	1805				
Taxes not included	327				
Total	7150	7150	7150	7150	7150
Total Cost	$11431	$11902	$11737	$11776	$11848
			PURCHASE PRICE		
Original Vehicle Cost	3185	4344	4074	3824	3745
Increase over 1970 ICE	0	1159	889	639	560
% Increase	0	36.4	27.9	20.1	17.6

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*Engine costs are for PD-1 and PD-2 engines; they are assumed to be unchanged for the PD-1A and PD-2A engines.

Table 38 (Cont'd.)

Comparison Summary
Weight

Item					
Engine Type	1970 ICE	CD-1	CD-2	PD-1	PD-2
		ENGINE WEIGHT (LBS)			
Basic Engine Weight	600	594	655	484	425
Accessory Weight	150	38	38	38	38
Transmission Weight	150	120	73	120	150
Total	900	752	766	642	613

ICE cost of ownership. The lower maintenance and oil costs help to balance the increased initial engine cost increment.

The comparison of original purchase price shows the increased cost of the turbine engine of between $560 and $1159. Since the basis of comparisons is a 1970 ICE, and since emission control devices may cost between $240 and $500 the actual increased cost in the 1975-1976 time period would be between $60 and $909.

In all cases the turbine engines are lighter than the ICE. This weight saving could undoubtedly be used to save vehicle cost.

3.8 Recommended Configuration

As indicated in the introduction one purpose of the present contract is to define the optimum gas turbine engine(s) capable of meeting the 1976 Federal Standards on automobile emissions and capable of being developed by the year 1975. The indications from reference 1 are that compromises in regenerated gas turbine engine cycles are required so that the combustors can meet the 1976 emission standards especially the standard on NO_x. By direction of EPA the Preliminary Design studies were to be directed toward two engine designs which facilitated the design of low-NO_x combustors. Two approaches were evolved for the low NO_x combustor. In the first, air for the primary zone of the combustor is bypassed around the regenerator, resulting in air at compressor discharge temperature. This moderate-temperature air reduces the primary-zone combustion temperature below the value which would result if regenerated air were used. The main benefit of this is the preclusion of premature ignition of the premixed fuel-air charge as it approaches the combustor. The premixed charge is then burned in a special combustor which has additional features chosen to limit NO_x production.

The second approach involves limiting the combustor inlet temperature to 1000°F instead of the usual 1200 to 1450°F obtained from a regenerator. The lower temperature also reduces the combustion temperature at a given fuel-air ratio and thus limits the formation of NO_x.

In the section (3.4) on design, two engines were optimized and designed, one for each of the above combustor design approaches. The regenerator bypass engine (PD-1A)designed for the General Electric low NO_x combustor is a regenerated single-shaft engine with variable stators in both the centrifugal compressor and single-stage axial turbine. The rated turbine inlet temperature design pressure ratio and power are 1900°F, 3.2 and 134 HP (105°F day), respectively. The 1000°F combustion inlet temperature engine (PD-2A) is a free-turbine engine with variable stators on the single-stage axial power turbine. The rated turbine inlet temperature, pressure ratio and power are 1900°F, 6.6 and 134 HP (105°F day), respectively.

3.8.1 Comparative Data

The basis of comparison is the net cost of ownership of a "standard" six-passenger automobile powered by a gas turbine engine over a ten year life including first cost minus scrap value, fuel cost, maintenance cost and repair cost. The automobile is to be driven 105,200 miles during the ten-year lifetime. The fuel cost is to be determined on the Uniform Simplified Engine Duty Cycle of Table 17 which includes the Federal Driving Cycle (reference 21) and suburban and country driving. Comparative data for the two engines are shown in Table 39. The PD-2A engine is lighter in weight but has a poorer average fuel mileage value. The PD-1A engine fuel economy is greatly aided by variable stators in both the compressor and turbine. As a result of the poorer fuel mileage of the PD-2A engine the fuel cost for the life of the automobile is $244 more than for the PD-1A engine.

The net cost of ownership is itemized in Table 40. The engine first cost is smaller for the PD-2A engine by $79. Also the maintenance cost is less for the PD-2A engine by $100 because there are no regenerator seals to be replaced. However, the fuel costs are less for the PD-1A engine by $244. Combining the lower fuel costs and higher salvage value (due to more weight of nickel bearing alloys) of the PD-1A a net saving of $72 under the PD-2A is realized on the net cost of ownership. This cost difference, of course, is within the overall accuracy of the cost figures. Therefore, the two engine powered automobiles can be said to have substantially the same net cost of ownership.

3.8.2 Comparison of Engine Features

The following are the advantages and disadvantages of the PD-1A engine:

	Advantages		Disadvantages
1.	Variable stators on compressor and turbine permit low-pressure-ratio compressor with a high low-power efficiency potential.	1.	The tendency for somewhat higher weight is inherent in the combined use of the regenerator and variable turbomachinery geometry.

Table 39

Comparative Data on Preliminary Design Engines

Designation	Type	Power, HP	Overall Fuel Mileage, mpg	Fuel Cost, $	Weight lb
PD-1A	Single Shaft	134	13.59	2391	642
PD-2A	Free Turbine	134	12.34	2635	610

Table 40

Comparative Net Cost of Ownership Data

Items of Cost	PD-1A	PD-2A
Engine	1,539	1,460
Salvage Value	-14	-7
Subtotal	1,525	1,453
Fuel	2,391	2,635
Subtotal	3,916	4,088
Vehicle	7,150	7,150
Maintenance and Repair	670	570
Lube Oil	40	40
Total	11,776	11,848

Advantages	Disadvantages
2. Low combustor inlet pressure cycle is conductive to low NO_x production.	
3. Use of the GE low NO_x combustor concept which fits this engine well has a high potential of meeting the 1976 Federal emission standards.	2. Use of regenerator by-pass required for low NO_x combustor penalizes fuel economy.
4. Almost no time lag on acceleration demand because opening stators and increasing fuel flow gives immediate increase in torque.	
5. Commercial-grade ball bearings are moderate in cost, reliable and enhance idle fuel economy.	3. The single shaft engine requires the development of an infinitely variable transmission complete with neutral and reversing mechanism.

The following are the advantages and disadvantages of the PD-2A engine:

Advantages	Disadvantages
1. Limiting combustor inlet temperature to 1000°F may ease the problem of obtaining low NO_x combustor.	1. The compromise in fuel economy is greater than for the GE low NO_x combustor, and the potential of meeting the NO_x emission standard is not as great.
2. The recuperator has less leakage and is less complicated than the regenerator while giving adequate effectiveness.	2. The transient engine performance is sensitive to rotor inertia, resulting in high gas-generator speed, high bearing losses and a lower pressure ratio than optimum from fuel economy standpoint.
3. The engine utilizes existing automatic transmission technology with torque converters.	3. The free turbine engine requires a second shaft, turbine and bearings, and enclosures and mounting provisions for them.
4. The minimum variable geometry simplifies the control.	

3.8.3 Recommendation

The regenerated, single-shaft(PD-1A) engine with variable primary combustor air regenerator bypass and with variable stators on compressor and turbine is recommended for development of the advanced automobile gas-turbine engine for the following reasons:

1. The PD-1A engine has no greater net cost of ownership than the PD-2A engine.
2. The General Electric low NO_x combustor has a higher potential for meeting the 1976 NO_x emission standard than the 1000°F inlet combustor.
3. The PD-1A engine permits considerable design freedom in developing the low NO_x combustor by virtue of the location of the combustor in the front of the engine and by virtue of the large volume available for the combustor.

3.9 Recommended Engine Development and Demonstration Program Plans

A program plan is set forth below for the technical demonstration of the single-shaft, variable-geometry, regenerated engine (PD-1A) selected in the previous section. The schedule for the development and demonstration is shown in Figure 121 and it is terminated with a demonstration of the engine at the end of calendar year 1975. In what follows estimates of manpower and funding requirements are delineated and critical timing and significant milestones are indicated for both the component development program and each phase of the engine development.

The engine development and demonstration program is divided into six tasks having various time spans, they are:

1. Establishment of an Engine Specification
2. Engine Preliminary Design
3. Component Development
4. Engine Final Design
5. Engine Hardware Procurement
6. Engine Demonstration

The objectives, and expected results are discussed below and the required resources are summarized in Table 41.

3.9.1 Establishment of an Engine Specification

In reference 20 is given the specifications for the performance of

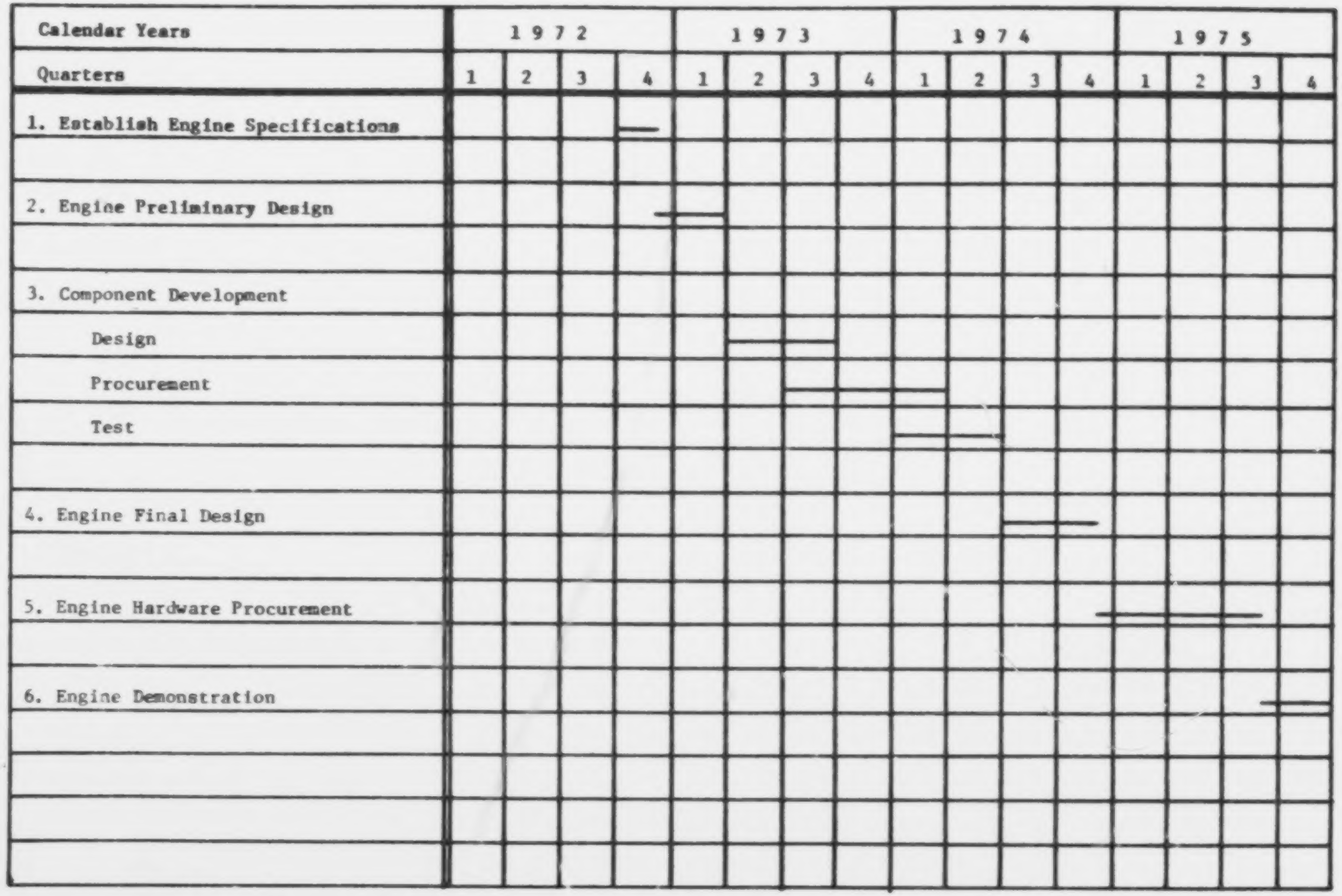

Figure 121. Gas Turbine Development Program

a passenger vehicle. It is the object of this task to write a specification for the gas turbine engine required to power the specified vehicle so that the vehicle will have the performance delineated. In the writing of this specification a number of things must be considered in addition to vehicle specification. For example, the vehicle and engine must comply with all Department of Transportation Motor Vehicle Safety Standards in force. In addition the specification must take into account engine operation under normal as well as extreme operating conditions, including starting, driving and stopping in ice, snow, rain, sleet, heat, dust, cold, mountains, city and country. Consideration must be given to minimizing the effects of tampering, abuse, "hot rodding", etc. Consideration must also be given to the effects of aging, inadequate maintenance and collisons. Noise and exhaust emission standards must also be factored into the specification.

In the writing of the engine specification, the anticipated state of the art after component development will be assumed. Target component performance will be established and the engine thermodynamic cycle will be established. Off design performance goals will also be set. The design criteria will be determined and the material to be used for each component will be delineated along with suitable alternates. Weight and cost targets will also be set.

The Establishment of an Engine Specification is estimated to require 3600 manhours over a period of 9 weeks. The task will require the expenditure of $100,800 including such material charges as computer services and Travel and Living expenses (See Table 41). At the end of this task an engine specification will be available from which a Preliminary Design of the engine can be made.

3.9.2 Engine Preliminary Design

The object of the Engine Preliminary Design is to carry out the design of the engine to the point at which the component development problems are clearly delineated and all major component sizes are established. In order to do this, the aerothermodynamic design of the gas flow path components will be carried out in parallel with the design of the mechanical components of the engine. Engine off design performance

Table 41

Automobile Gas Turbine Development Program - Costs

	Manhours	Labor	Material	Total Cost
1. Establishment of an Engine Specification (2 mos)	3,600	97,900	2,900	100,800
2. Engine Preliminary Design (4 mos)	6,800	184,700	9,200	193,900
3. Component Development (15 mos)				
3.1 Compressor Design	6,470	123,000	7,000	130,000
Procure	650	20,000	113,000	133,000
Facilities	2,420	36,000	110,000	146,000
Test	8,140	122,000	12,000	134,000
	17,680	301,000	242,000	543,000
3.2 Regenerator Design	3,380	90,000	6,000	96,000
Procure	480	15,000	84,000	99,000
Facilities	880	18,000	57,000	75,000
Test	10,190	214,000	21,000	235,000
	14,930	337,000	168,000	505,000
3.3 Combustor Design	4,150	107,000	7,000	114,000
Procure	1,320	41,000	226,000	267,000
Facilities	600	12,000	38,000	50,000
Test	3,820	78,000	8,000	86,000
	9,890	238,000	279,000	517,000
3.4 Turbine Design	10,180	277,000	14,000	291,000
Procure	6,290	195,000	823,000	1,018,000
Facilities	2,000	43,000	129,000	172,000
Test	23,700	510,000	37,000	547,000
	41,810	1,025,000	1,003,000	2,028,000

Table 41 (Cont'd.)

Automobile Gas Turbine Development Program - Costs

			Manhours	Labor	Material	Total Cost
3.5	Rotor, Bearings & Seals	Design	10,870	293,000	17,000	310,000
		Procure	770	24,000	136,000	160,000
		Facilities	2,880	62,000	185,000	247,000
		Test	11,860	255,000	25,000	280,000
			26,380	634,000	363,000	997,000
3.6	Transmission	Design	11,060	301,000	19,000	320,000
		Procure	1,350	42,000	238,000	280,000
		Facilities	1,120	24,000	74,000	98,000
		Test	9,300	200,000	20,000	220,000
			22,830	567,000	351,000	918,000
3.7	Control and Actuator	Design	11,030	300,000	18,000	318,000
		Procure	1,480	46,000	265,000	311,000
		Facilities	320	7,000	22,000	29,000
		Test	13,590	292,000	29,000	321,000
			26,420	645,000	334,000	979,000
3.8	Inlet & Exhaust	Design	620	16,900	1,100	18,000
		Procure	110	3,400	19,600	23,000
		Facilities	190	4,000	12,000	16,000
		Test	800	17,300	1,700	19,000
			1,720	41,600	34,400	76,000
3.9	Auxiliaries & Accessories	Design	2,180	59,200	3,800	63,000
		Procure	320	9,800	55,200	65,000
		Facilities	130	2,800	8,200	11,000
		Test	2,620	56,400	5,600	62,000
			5,250	128,200	72,800	201,000

Table 41 (Cont'd.)

Automobile Gas Turbine Development Program - Costs

		Manhours	Labor	Material	Total Cost
3.10 Production Manufacturing Methods	Design	15,720	428,000	29,000	457,000
	Procure	4,740	147,000	797,000	944,000
	Facilities	4,230	91,000	270,000	361,000
	Test	43,900	946,000	82,000	1,028,000
		68,590	1,612,000	1,178,000	2,790,000
4. Engine Final Design (6 mos)		13,520	367,500	22,000	389,500
5. Engine Hardware Procurement (9 mos)					
5.1 Seven (7) Engines		12,900	400,000	2,280,000	2,680,000
5.2 Four (4) Vehicles		90	2,700	18,000	20,000
5.3 Modify Four (4) Vehicles		870	27,000	3,000	30,000
		13,860	429,700	2,301,000	2,730,700
6. Test and Evaluation (4 mos)					
6.1 Engine Dynamometer		3,700	79,620	108,650	188,270
6.2 Chassis Dynamometer		19,670	423,150	345,690	768,840
6.3 Road Test		10,490	225,500	101,548	327,048
		33,860	728,270	555,888	1,284,158
Total		307,140	7,336,870	6,916,188	14,253,058

will be carried out to facilitate vehicle mission analyses and the design of the engine control. Design studies of the final drive, the inlet and exhaust systems, the auxiliaries and the accessory mounting provisions will permit engine configuration and vehicle installation studies to be carried out. The engine configuration studies eventually will be the basis for the thermal and stress maps of the engine on a steady state and transient basis. Failure mode and criticality analyses will be based on the configuration studies and the thermal and stress analyses. The most cost effective means of making the various engine parts will also stem from the configuration studies. Analyses will also be performed to establish the required amount and location of acoustic and thermal insulation. The culmination of this design work will be the determination of which engine specifications, including performance targets, can not be met. Component developments will then have to be carried out to attempt to meet the specification. The specifications for the design of the components needing development will also be available at this time.

The Engine Preliminary Design is estimated to require 6800 manhours over a period of 17 weeks. The task will require the expenditure of $193,900 including such material charges as computer services and travel and living expenses (See Table 41). The expected results include layout and installation drawings of the engine, a clear delineation of the several required component development programs and specifications and sizing from which detailed component designs can be made.

3.9.3 Component Development Programs

The following ten component development programs have been identified and are described below:

Compressor
Regenerator
Combustor
Turbine
Rotor, Bearings and Seals
Transmission
Engine Control and Actuators
Inlet and Exhaust Systems
Auxiliaries and Accessories
Production Manufacturing Methods

These programs each have design, procurement and testing subtasks. In addition, costs of test vehicle and/or facilities were estimated. The Component Development Programs are scheduled for a 15-month time period (see Figure 121). It is estimated that 235,500 manhours will be required for a total cost of $9,554,000.

Compressor - The objectives of the compressor development are:

- Determination of the highest compressor design pressure ratio at which the variable-stator compressor has adequate flow range and efficiency to meet the demands of the engine.
- Determination of the effect of stator-vane end clearance on compressor off design performance.

In PD-1A,the design pressure ratio was limited to 3.2 because it was felt that adequate compressor performance over the engine operating conditions could not be counted on above this pressure ratio. However, the engine performance continues to increase with pressure ratio up to 4:1 if the compressor efficiency does not fall off. In the development program, compressors will be designed at pressure ratios from 3.2 to 4.0 and tested with variable stators. From the evaluation of the test data the highest practical pressure ratio commensurate with engine requirements will be established. A by product will be an experimental compressor map at the selected pressure ratio. At the highest practical pressure ratio, the effect of vane clearance on performance will be investigated.

The compressor development program requires 17,680 manhours and the expenditure of $543,000 including the test vehicle and the compressor flow-path parts (see Table 41). The expected results include establishment of the highest practical design pressure ratio, data on the effect of stator-vane end clearance and an experimental compressor performance map for a compressor sized for the engine.

Regenerator - The objectives of the regenerator development are:

- Determine the best sealing materials for the regenerator.
- Develop an adequate regenerator sealing mechanism.

In this program seal wear rates would be determined for a number of candidate coating-substrate materials combination over a range of contact pressures and temperatures. The material variables to be investigated include composition,method of application,density (% of theoretical), thickness, post application treatments and substrate material. Full scale seals would then be fabricated from the material(s) and by the process(es) found to have the greatest potential for long life service and sealing tests made of ceramic-core regenerators to demonstrate the effectiveness of the sealing material.

The regenerator development program requires 14,930 manhours and the expenditure of $505,000, including test vehicle and facilities (see Table 41). It is expected that the specifications for a coating-substrate pair would evolve along with a suitable sealing mechanism. Wear and leakage rates would be obtained experimentally for the selected sealing mechanism.

Combustor - The objectives of the combustor development are:

- Evolve a configuration which has lower emission levels than the established standards.
- Develop a fuel-air mixing and vaporization system.
- Demonstrate acceptable emission levels during steady state and transient operation.

In this program, scale model tests of several configurations would be run to establish the design which gave the best compromise between quickly quenching the NO_x and providing enough residence time to burn the CO and HC. In parallel a fuel-air mixing system would be developed which gives a uniform fuel-air mixture. A full size combustor with vaporizer would then be tested on a typical automotive fuel, and alternate fuels.

The combustor development requires 9,890 manhours and the expenditure of $517,000, including a test vehicle and facilities (see Table 41). It is expected that a combustor would be demonstrated which had emission levels significantly below the standard when operated on automotive gas turbine fuel.

Turbine - The objectives of the turbine development program are:

- Develop a low-cost variable-stator turbine sized for the engine and having high efficiency over the operating range of the engine.

- Develop low-cost turbine stator vanes with adequate temperature capability.
- Develop low-cost turbine static hot parts.
- Develop low-cost, high-temperature turbine wheel.

A variable-stator mechanism will be designed, minimizing leakage, back lash and actuation force, fabricated and tested. With the object of saving production cost variable stator blades and other hot static parts will be developed using hot-pressed or reaction bonded silicon nitride or silicon carbide. After selection of the material having the greatest potential of meeting the requirements for low-cost, various manufacturing processes will be investigated to evaluate the impact on cost effectiveness and requisite materials properties. Full-scale hardware will be fabricated and tested. Means for reducing tip seal cost will be pursued by investigating abradable spray coatings and the possibility of reducing blade thickness ("squealer tips") near the casing to prevent damage in the event of a turbine wheel rub. Reducing turbine-wheel cost will be investigated by making the hub from a lower cost alloy than the rim and integral blades followed by electron beam or inertia welding. This will be compared with casting the wheel from IN 713LC. The wheel will be made by the cheaper process. The turbine will be fabricated from the materials identified during this development program. The wheel will be destructively tested in a spin pit at temperature and the entire turbine will be tested at temperatures for operational characteristics and performance.

The turbine development program requires 41,810 manhours and the expenditure of $2,028,000, including the test vehicle (see Table 41). It is expected that a high-efficiency turbine suitable for an engine will be demonstrated which will be amenable to low-cost production. Off design performance will be measured and rotor strength will be demonstrated.

Rotor, Bearings and Seals - The objectives of Rotor, Bearings and Seal program are:

- Develop bearings permitting stable operation at high rotative speeds with low losses and low cost.
- Develop shaft, bearing and seal arrangement which minimizes vibrational difficulties.

- Develop adequate lubricating system including the required seals.

Considered in the bearing development program will be low-cost gas, ball or film bearings, in the latter case including "slipper" bearings. Complete rotor-bearing systems will be designed and at least one will be selected for fabrication and test. Vibration analyses of the rotor-bearing combination using the several bearing types will be carried out as part of the selection process. Bearing stability analyses will also be required. The lubrication system of the type needed for the selected bearing type will be designed. The selected rotor-bearing system, including the lubrication system will be fabricated and tested with appropriate masses simulating the turbomachinery so that vibrational and stability characteristics can be measured. Appropriate modifications will be made in the rotor bearing system based upon the test result so that the system will fulfill the requirements of the engine.

The Rotor, Bearing and Seal program will require 26,380 manhours and will require the expenditure of $997,000 (see Table 41). The expected result is a rotor,bearing and seal system permitting the high rotative speeds comensurate with low inertia for rapid acceleration of the rotor, low loss comensurate with low idle power loss and low cost.

Transmission - The objectives of the Transmission development program are:

- Development of an infinitely variable transmission with the following characteristics.
  - Continuously variable from high forward speed through neutral to moderate speed in reverse.
  - Containing a built-in reduction gear to normal drive shaft speeds.
  - Automatically controlled.

Considered in the program will be various means of providing a neutral position to eliminate the need for a slipping clutch (since the single-shaft engine cannot run at zero speed). The design will be refined so as to contain the appropriate reduction gearing from engine speed to drive-shaft speed and the appropriate control for the transmission will be developed. The entire transmission will be tested in a dynamometer so that the adequacy of the design can be experimentally assessed and so that performance can be measured.

The Transmission program will require 22,830 manhours and the expenditure of $918,000, including the test vehicle and dynamometer (see Table 41). It is expected that the program will be able to demonstrate a completely automatic transmission with forward neutral and reverse. Transmission efficiency data will also be measured.

Control and Actuator - The objectives of the Control and Actuator development program are:

- Develop an engine control so that the engine can be completely controlled by one control lever and a selector quadrant, giving due consideration to cost.
- Develop an actuator system best suited to the engine and control, giving due consideration to cost.

Transient engine performance analyses to meet the requirements of the vehicle (reference 20) will be carried out to establish the control philosophy, Control devices comensurate with the established control philosophy will be identified and a control will be designed using the devices which provide adequate control and have the potential of low cost. A "bread board" control will be built and tested on a computerized engine simulator.

The Control and Actuator program will require 26,420 manhours and the expenditure of $979,000 (see Table 41). It is expected that a "bread-board" control which meets all of the vehicle performance requirements of reference 20 can be demonstrated in an engine simulator. Specific control performance data needed in engine design will be obtained experimentally.

Inlet and Exhaust Systems - The objectives of the Inlet and Exhaust Systems program are:

- Develop inlet and exhaust systems which have the following characteristics:

- Low flow losses
- Prevent entrance of dust, ice, snow, rain, hail, salt and foreign objects
- Provision of take off for passenger compartment heating
- Suitable for installation in the automobile

Configuration studies of inlet and exhaust systems installed in the vehicle will be made having provisions for an air filter and heating of the passenger compartment. A selected design on the basis of function and cost will be designed and subjected to tests.

The Inlet and Exhaust Systems program will require 1720 manhours and require the expenditure of $76,000 (see Table 41). The results expected are test data on systems suitable for vehicle installation, including pressure drop, heat exchanger capacity and filtration.

Auxiliaries and Accessories - The objectives of the Auxiliaries and Accessory program are:

- Determine which of the auxiliaries are available on the market.
- Establish specifications for non-standard auxiliaries and procure two of each for evaluation.
- Translate all accessory requirements into space requirements on the engine preliminary design drawing.

Specifications for non-standard auxiliaries will be drawn up by reference to similar standard items. Suitable vendors for the non-standard auxiliaries will be identified and the auxiliaries procured. Appropriate evaluation tests will be performed on the procured auxiliaries. With regard to accessories, provisions will be made on the engine drawing for the use of standard automobile accessories unless this unduly compromises the engine design.

The Auxiliaries and Accessories program will require 5250 manhours and require the expenditure of $201,000 (see Table 41). The expected results are two each of all non-standard auxiliaries and provisions on the engine drawing for all required accessories. The procured auxiliaries will be subjected to qualification tests.

Production Manufacturing Methods - The objectives of the Production Manufacturing Methods program are:

- Identify and incorporate on the drawings design changes which reduce the cost of making the several engine parts.
- Identify the processes for making each engine part.
- Where alternate or untried processes are potentially lower in cost make parts on experimental (non-production) machines.
- Determine the equipment investment required to mass produce the engine.
- Determine the labor cost of mass producing the engine.

A study of the engine preliminary design will be made to simplify the design and improve its manufacturability. Where possible the number of parts will be reduced in cases in which one part will perform the function of two or more parts. Complicated parts will be divided into more parts when such division results in a potential cost saving. The engine drawings will be updated to incorporate the design changes. The alternate ways of making parts will be studied with the view toward finding methods giving the least cost. In the compressor low-cost aluminum casting alloy must be identified which has adequate fatigue strength to prevent blade vibration failures. Test specimens of several casting alloys will be cast for complete characterization of these materials including determination of fatigue properties over a range of temperatures, cycles, ratios of alternating to mean stress and stress concentrations. Other materials characteristics that must be evaluated in addition to mechanical properties are salt spray corrosion, stress corrosion and thermal stability. Alternate methods of making turbine wheels include ceramics and powder metallurgy. Each method will be investigated for feasibility and to assess the probability on making a lower cost wheel. Materials most suitable for each fabrication process will be screened and at least one material selected for each process development. Full scale rotors will be produced by the identified methods and subjected to evaluation by physical tests, including spin-pit tests at elevated temperatures. The selected material and process will be subjected to process optimization studies to determine the most cost effective means of making the wheels. The candidate materials for the low-NO_x combustor are a number of grades of structural ceramics. Two or more materials will evaluated to determine the most cost effective means of making the combustor. Evaluation will include sufficient process development to achieve satisfactory structures for test. Sufficient characterization tests will be conducted on specimens representative of the various commercial manufacturing techniques as to permit the selection of a suitable material and commercial process. Subsequently the processing procedures will be optimized and scaled up to yield full scale pieces of proper dimensions, structure and properties. As in the case of the major materials developments, alternate materials or processes with a high potential of cost saving in other parts of the engine will be investigated. Parts made of such materials or by such processes will be fabricated and evaluated by destructive or non-destructive means as appropriate to determine whether the parts meet the specifications. The equipment requirements for mass producing the engine by the selected methods will be ascertained and the cost of the equipment will be estimated. The labor cost of mass producing the engine will also be estimated.

The Production Manufacturing Methods program will require 68,590 manhours and the expenditure of $2,790,000 (see Table 41). The expected results include the most economical methods for producing the engine along with the cost estimates of production machinery and labor per engine produced.

3.9.4 Engine Final Design

The objective of the Final Design is manufacturing drawings and specifications for the fabrication of the seven engines needed in the engine development program. The impact of missed goals in the development program first will be assessed and translated into changes in the engine specifications. Based upon the new specifidations the aerothermodynamic design of the flow-path parts will be carried out closely followed by the design of the mechanical components of the engine. Off design performance of the engine will be determined so that vehicle performance can be compared with the requirements of reference 20. The off design performance will also be used to design the engine control. The auxiliaries will then be designed and provisions will be made for the usual accessories. The design of the inlet and exhaust system and the power train, including the transmission, will permit vehicle installation studies to be made. The engine configuration layout will permit thermal and stress analyses to be performed. These analyses will be used in materials choice decisions and in failure mode and criticality analyses. As the design proceeds the most cost effective methods for making the several engine parts identified during the Component Development program will be factored into the engine design. The engine design will include acoustic and thermal insulation. The engine design will be completed with manufacturing drawings and specifications for the seven engines required by the Program Plan. Drawings and specifications will also have to be made for the modification of stock automobiles so that gas-turbine engines can be installed.

The Engine Final Design will require 13,520 manhours over a time period of 6 months and will require the expenditure of $389,500 (see Table 41). The results will include manufacturing drawings and specifications for an engine based upon an extensive Component Development program and the concommittant modification designs of stock automobiles in which the engines are to be installed.

3.9.5 Engine Hardware Procurement

Hardware will be procured for the assembly of seven engines. The seventh engine will be a spare. Also four stock automobiles will be procured and modified to accept the procured gas-turbine engines. This task includes procurement, expediting, quality control and material review board activities. Included also is engine assembly and automobile modification. This task will require 13,860 manhours over a 9-month period and the expenditure of $2,730,700 (see Table 41). The results expected include the procurement of parts for and assembly of seven engines and the procurement and modification of four vehicles.

3.9.6 Engine Test and Evaluation

The objectives of the Engine Test and Evaluation are:

- Determine the steady state and transient performance of the engine.
- Determine the emission levels of the engine.
- Determine the controllability of the engine.

- Determine the performance of the vehicle powered by the engine.
- Determine the noise and vibration characteristics of the engine when installed in the vehicle.

Seven engines are required for the Engine Test and Evaluation program, three engines, two of them eventually installed in vehicles, will be evaluated by the contractor and the same number by EPA. The seventh engine is a spare to minimize program delays by engine malfunction. Shown in Figure 122 is the schedule of the contractor Engine Test and Evaluation program. The first engine will be set up and checked out in an engine dynamometer. Control studies will follow, permitting the proper adjustment of the controls and an evaluation of control operation. Engine performance tests will follow control studies and following these will be emission level tests run on the engine dynamometer. Next, the first engine will be installed in a vehicle which will be placed in a chassis dynamometer. Performance tests on the engine, both steady-state and transient, will follow. These tests, like the emission tests to follow, will be run with the engine programmed to follow the Federal Driving Cycle (FDC). Next engine and vehicle performance tests will be carried out on the test track. During road testing noise and vibration tests will be carried out and performance data will be gathered at various engine operating temperature levels.

The second engine will be installed in a vehicle immediately and the vehicle will be placed on a chassis dynamometer to measure engine performance and exhaust emissions. Next the vehicle will undergo evaluation on a test track in which controllability will be studied along with engine and vehicle performance. Six months of endurance testing will follow after which the vehicle will be returned to the chassis dynamometer to ascertain the effect of endurance on exhaust emissions and engine performance.

The third engine will be installed in a vehicle immediately and placed on a chassis dynamometer. Thirteen months of endurance testing will take place during which the FDC, suburban and country routes will be simulated.

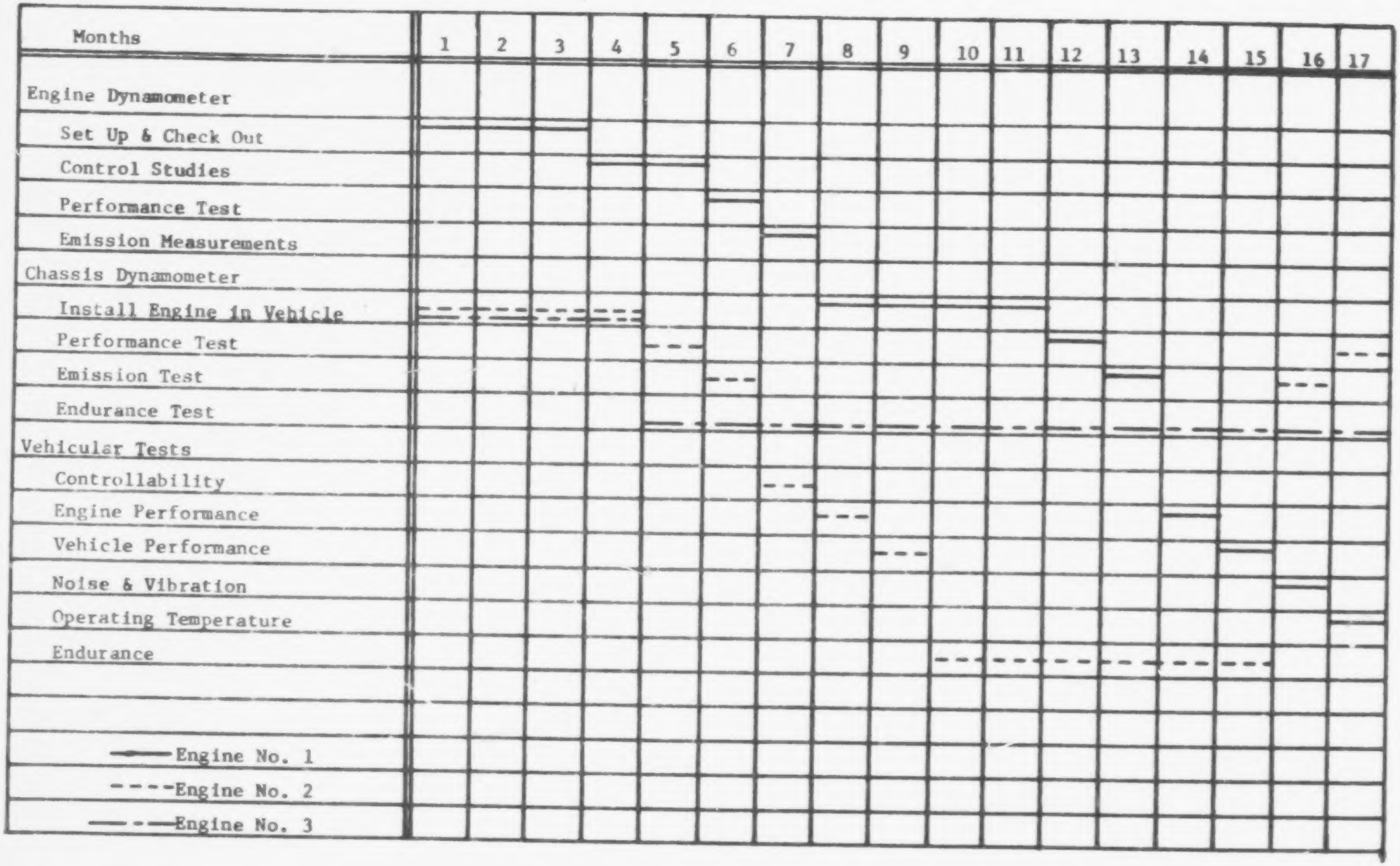

Figure 122. Gas Turbine Engine Demonstration Program

The Engine Test and Evaluation program will require 33,860 manhours and $1,284,158 (see Table 41). The expected results include a demonstration that the developed engine is suitable for automotive use and test data on engine performance, exhaust emissions, engine controllability, gas-turbine-powered vehicle performance and noise and vibration characteristics when installed in a vehicle will be obtained.

4.0 CONCLUDING REMARKS

After screening ten gas turbine cycle types three were selected for Conceptual Design Studies, they were:

- CD-1 Regenerated, single shaft engine with a variable compressor and turbine.
- CD-2 Regenerated, free-turbine engine with a variable power turbine.
- CD-3 Regenerated, free-turbine engine with the compressor and both turbines variable.

Layout drawings were made of all three engines but the CD-3 engine was eliminated from further study because the small improvement in low power fuel economy did not justify the added complexity. The following results were obtained for the CD-1 and CD-2 engines.

The CD-1 engine weighed 752 lbs. complete with transmission, had a design pressure ratio of 3.6:1 and it was found that this engine could meet the vehicle acceleration requirement with a rated power of 134 HP. The mission analysis indicated that the average fuel mileage was 16.06 mpg and that the required fuel cost for the life of the engine (10 years, 105,200 miles) was $2025. The economic analysis revealed that the engine first cost was $2059 and that the net cost of ownership for the engine and automobile was $11,902.

The CD-2 engine weighed 766 lbs. complete with transmission, had a design pressure ratio of 5.0:1 and it was found that this engine could meet the vehicle acceleration requirements with a rated power of 150 HP. The mission analysis indicated that the average fuel mileage was 15.39 mpg and that the required fuel cost for the life of the engine (10 years, 105,200 miles) was $2114. The economic analysis revealed that the engine first cost was $1789 and that the net cost of ownership for the engine and automobile was $11,737.

The CD-1 and CD-2 engines had no special provisions which would facilitate the design of a combustor which could meet the 1976 Federal standards on NO_x production. However, early test results presented at the January Automotive Power Systems Contractors' Coordination Meeting sponsored by EPA indicated that the inlet conditions to the combustor on a regenerative engine had to be controlled if the standard were to be met. As a result two combustor design concepts were adopted and preliminary designs, off-design performance analyses, mission analyses and economic analyses were made on two engines, each designed for a different combustor concept. The studies of these two engines drew heavily upon the results obtained for the conceptual design engines and many refinements were incorporated into the preliminary design engines. The following results were obtained for the two engines with special provisions to lower the NO_x emission of the combustors.

The PD-1A engine is a regenerative, single shaft engine with variable stators in the compressor and turbine and is provided with a variable bypass around the regenerator for the combustor primary air. The GE low-NO_x combustor was one of the design features of this engine. This engine weighed 642 lbs. complete with transmission, had a design pressure ratio of 3.2:1 and it was found that this engine could meet the vehicle acceleration requirement with a rated power of 134 HP. The mission analysis indicated that the average fuel mileage was 13.59 mpg and that the required fuel cost for the life of the engine (10 years, 105,200 miles) was $2391. The economic analysis revealed that the engine first cost was $1539 and that the net cost of ownership for the engine and automobile was $11,776.

The PD-2A engine is a recuperated, free turbine engine with a variable power turbine and was designed so as to limit the combustor inlet temperature to 1000°F under all operating conditions. This engine weighed 613 lbs. complete with transmission, had a design pressure ratio of 6.6:1 and it was found that this engine could meet the vehicle acceleration requirements with a rated power of 134 HP. The mission analysis indicated that the average fuel mileage was 12.34 mpg and that the required fuel cost for the life of the engine (10 years, 105,200 miles) was $2635. The economic analysis revealed that the engine first cost was $1460 and that the net cost of ownership for the engine and automobile was $11,848.

The single-shaft engine with the regenerator bypass for the primary combustor air was recommended for the advanced automobile gas turbine engine because the GE low-NO_x combustor, which has a higher potential of meeting the NO_x standard than the 1000°F combustor, is completely compatible with this engine and the net cost of ownership of this engine was no higher than for the free-turbine engine with the 1000°F combustor. The plan for the development and demonstration of this engine within 1975 was delineated, including component development, design, procurement and testing of the engine on dynamometers and in automobiles.

5.0 ACKNOWLEDGEMENT

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6.0 CONTRACT TASK STRUCTURE

The contract was carried out in nine tasks. The major contributors from the General Electric Company are shown below under the appropriate tasks.

Establish Baseline Technology

C.W. Deane
A.W. Schnacke
G.C. Wesling

Parametric Design Point Cycle Study

C.S. Robertson
A.W. Schnacke
G.C. Wesling

Preliminary Candidate Cycle Selection

R.J. Rossbach
A.W. Schnacke

Off-Design Performance Analysis

R.L. Henderson
G.C. Wesling

Mission Analysis

D.H. Brown
C.W. Deane
T.L. Schilling

Design

A.W. Schnacke
G.C. Wesling

Economic Analysis

D.E. Dutt
C.S. Robertson

Recommended Configuration(s)

R.J. Rossbach
A.W. Schnacke

Program Plans - Recommended Engine Development and Demonstration

R.G. Frank
R.J. Rossbach

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23. Final Report of the Ad Hoc Committee, "Cumulative Regulator Effects on the Cost of Automotive Transportation (RECAT)," Feb. 28, 1972, Office of Science and Technology.

Appendix A - Economic Analysis

Sample Process Sheets, MES Analysis - As discussed in Section 3.7.1.1, the costing of the direct labor and materials for each engine was performed by Manufacturing Engineering Services, (MES), a component of the General Electric specializing in all aspects of mass production. This Appendix contains samples of the process sheets prepared by MES, and indicate the level of detail used to arrive at the direct manufacturing cost.

Table A-1 shows the material costing for nine parts of CD-1 engine. NSP supplied the descriptions, material, volume, and approximate weight of the finished part. MES selected the manufacturing process (such as casting or bar stock), estimated the cost per pound from vendor quotes or prior experience, and calculated the finished part material cost. In some cases, such as part 4, MES broke the part into two or more subparts for costing purposes. Table A-2 shows the technique used to account for chips, scrap, salvage, etc. The total cost and weight of each class of material (as shown in Table A-2) were determined, and the percent adder applied to get the scrap cost. Purchased parts and items such as nuts, bolts, bearings, seals, etc. were costed separately. For the two conceptual design engines, it is likely that many of this type of part were left out, however, the analysis of the two preliminary design engines showed that the error thus introduced was very small.

The steps required to finish each part were estimated as shown on Table A-3. The number of machines and men required to produce 5000 units per day was calculated from the selected procedure, and the man power converted to dollars per unit as shown in Table A-4. The very low per unit labor costs are due to the use of specialized machinery tailored to the part to be processed.

Table A-5 shows the assembly sequence with the time in seconds required to complete each operation. The total of these assembly times gave the total assembly cost as shown in Section 3.7.

Automotive Gas Turbine Single Shaft Type (CD-1)
[illegible]

Table A-1

Material Cost Sheet

No.	Part Name	Material	Approximate Size	Volume in In^3	Approx. Wt. in lbs.	Kind of Raw Mat'l	Material Cost/Lb.	Casting Cost/Lb.	Mat'l/Cast. Cost/Piece	No. of Parts/Unit	Mat'l or Cast Cost Per Unit
1	Main Housing	Nodular Iron	20" x 21-3/4" x 22-1/2" Wall thickness varies from 1/4" to 1"	562.12	146.15	Casting		$ .30	$43.85	1	$43.85
2	Regenerator Cover Rt. Side	Nodular Iron	3 1/4" x 21 3/4" x 24 1/4" Wall Thickness varies from 3/16" to 1"	97.47	25.34	Casting		.30	7.60	1	7.60
3	Regenerator Cover Left Side	Nodular Iron	3 1/4" x 21 3/4" x 24 1/4" Wall thickness varies from 3/16" to 1"	97.47	25.34	Casting		.30	7.60	1	7.60
4	A - Turbine Rotor	Inc. 738	9 1/2" dia x 2 1/4" Depth	55. 0	16.5	Casting		5.50	90.75	1	90.75
	B - Turbine Rotor Shaft	Shaft Steel	1 1/4" dia x 12 1/2" Long	14.63	4.1	Bar Stock	$ .20		.82	1	.82
5	Compressor Rotor	Alum.	8" dia. x 3" deep	22.43	2.24 Add .6 for blades Total Wt. 2.84	Casting		9.50	21.28	1	21.28
6	Compressor Rotor Shroud	Nodular Iron	18 ½ dia. x max. depth 7½" wall thickness ¼" to 5/8"	86.16	22.40	Casting		.40	8.96	1	8.96
7	Compressor Front Shroud - Outer?	Nodular Iron	19½" dia. x max. depth 3 3/4" wall thickness 3/16" to 5/8"	103.45 + 29.45 for pin support volume	34.55	Casting		.45	15.55	1	15.55
8	Compressor Front Shroud - Inner ?	Nodular Iron	23 5/8" x 28" x 7½" max. depth wall thickness ¼" - 3/8"	207.75	54.01	Casting		.35	18.90	1	18.90
9	Gear Case	Nodular Iron	20 3/4" x 28" x 2 7/8" max. depth wall thickness ½"- 5/8"	121.22	31.52	Casting		.30	9.46	1	9.46

Table A-2

ADDER SHEET FOR CHIPS, SALVAGE, ETC. ON CASTINGS AND MATERIALS

	% Adder	$ Adder
Castings	10%	$ 37.50
Forgings	20%	1.96
Sheet Metal	30%	38.52
Bar Stock and Extrusions	10%	4.69
		$ 82.67

Table A-3

Manufacturing

	Part# and Name	Operation	Equipment	Men/Day
#1	Main Housing Nodular Iron Casting	1. Qualify-Blanchard Grind One surface for location	Blanchard Grinder	9
		2. Turn, Face, Bore, c/Bore Groove, Drill and tap Front & Rear Face of Casting. 3. Reposition on front face and turn, face, bore, c/bore, drill and tap both sides of casting.	Multi Station Transfer Machine	18
#2	Regenerator Cover Right Side Nodular Iron Casting	1. Chuck on O.D. and face turn and face rabbit fit, bore, c/bore, and groove.	4 Spindle Vertical	9
		2. Face, c/bore, drill and tap. 3. Face, drill & tap gear case drive. 4. Face, Bore, drill, tap, & c/bore side of gear case drive.	Multi Station Transfer Machine	9
#3	Regenerator Cover Left side Nodular Iron Casting	Same as Part #2		18

Table A-4

MANUFACTURING

Single Shaft Gas Turbine Engine #221R903 (CD-1)
All Costs Based On Production of 1,000,000 per year or 5,000/day
Direct Labor Cost At $5.00/Hour

Part # Name	Man Days Per Day	Dollars Each	No. Parts Per Unit	Dollar Per Un
1-Main Housing	27	$ .216	1	$ .216
2-Regenerator Cover Rt.side	18	.144	1	.144
3-Regenerator Cover Left Side	18	.144	1	.144
4-Turbine Rotor	42	.336	1	.336
5-Compressor Rotor	9	.072	1	.072
6-Compressor Rear Shroud	36	.288	1	.288
7-Compressor Front Shroud	36	.288	1	.288
8-Compressor Front Shroud-Inner	18	.144	1	.144
9-Gear Case	15	.120	1	.120
10-Turbine Scroll Shroud	82	.656	1	.656
1-Turbine Stator-Inner Shroud	30	.240	1	.240
2-Turbine Shaft Bearing Mt.	9	.072	1	.072
3-Turbine Shaft Seal	1½	.012	1	.012
4-Turbine Shaft Spacer	3½	.028	1	.028
5-Turbine Shaft Spacer	3½	.028	1	.028
6-Turbine Shaft Spacer	3½	.028	1	.028
7-Labrinth Seal	6	.048	1	.048
8- Turbine Tip Seal	15	.120	1	.120
9-Turbine Diffuser-Outer	17	.136	1	.048
0-Turbine Diffuser-Inner	39	.312	1	.312
sulation Shield	12	.096	1	.096

Table A-5

ASSEMBLY

Scroll Shroud Assembly

Time in Sec.	
88	Asm Pt. 92, guide buttons to Pt. 40, turbine nozzle Actuator ring.
6	Position Pt. 40, actuator ring on Pt. 10, scroll shroud
-36	Asm. Pt. 38, vanes, Pt. 37 nozzle adj. arms & Pt. 39, nozzle locknuts to Pt. 10, scroll shroud
5	Asm. Pt. 110, piston rings to Pt. 6, compressor rear shroud
60	Asm. Pt. 11, turbine stator inner shroud to Pt. 10, scroll shroud
20	Asm. Pt. 10, scroll shroud sub asm. to Pt. 6, compressor rear shroud
4	Asm. Pt. 17, Labyrinth seal to Pt. 4, rotor
5	Asm. Pt. 4, rotor into Pt. 6 compressor rear shroud
18	Invert Pt. 6, compressor rear shroud & position Pt.109, oil seal and Pt. 16, spacer, Pt. 108, bearing & Pt. 15, spacer onto Pt. 4, rotor
8	Asm. Pt. 108, bearing into Pt. 12, bearing mount & fasten with Pt. 13, seal mount
18	Bolt Pt. 12, sub asm. to Pt. 6, compressor rear shroud
5	Asm. Pt. 14, spacer & Pt. 107 oil seal on Pt. 4 rotor
60	Insert Pt. 18, tip seal sub asm. into Pt. 10, shroud sub asm. & bolt Pt. 19, outer diffuser to Pt. 10, shroud sub asm.
50	Position Pt. 111, piston rings into Pt. 1, main housing. Bolt Pt. 21 insulation shield to Pt. 1, main housing
70	Position and bolt Pt. 6, compressor rear shroud sub asm. into Pt. 1, main housing
18	Place Pt. 112 key & Pt. 5, compressor rotor on Pt. 4, turbine rotor & fasten with Pt. 44, compressor lock nut
915 Sec.	

Balloon Drawings and Parts Lists, PD-1 Engine - Figures A-1 thru A-3 (Dwg. 221R908) were prepared with each part number identified. Some details are not exactly the same as those shown in Dwg. 221R910, the final set, but the differences, such as component weight, were factored into the economic analysis.

Table A-6 is a list of the detailed parts lists prepared from the results of the MES study and in-house work. Note that the computer method used to prepare these tables can select any possible combination on order for other types of reports. The set included shows the breakdowns which were most useful for identifying potential cost reduction areas.

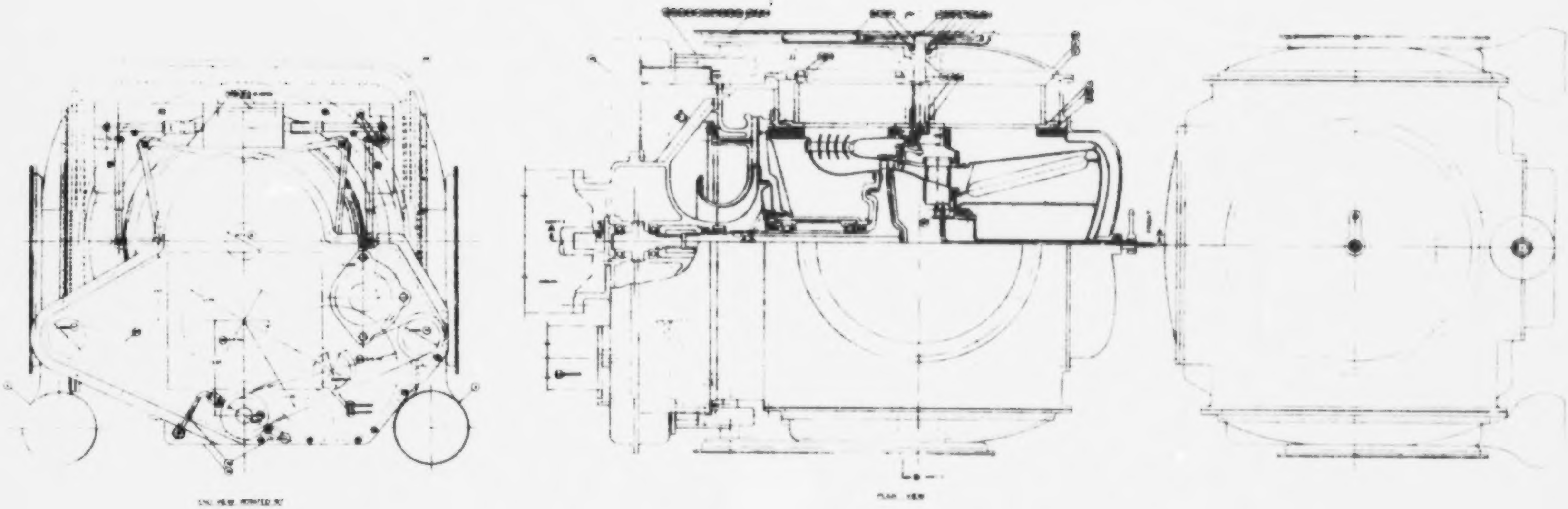

Figure A-1. PD-1 Engine Drawing With Part Numbers. Plan and End Views.

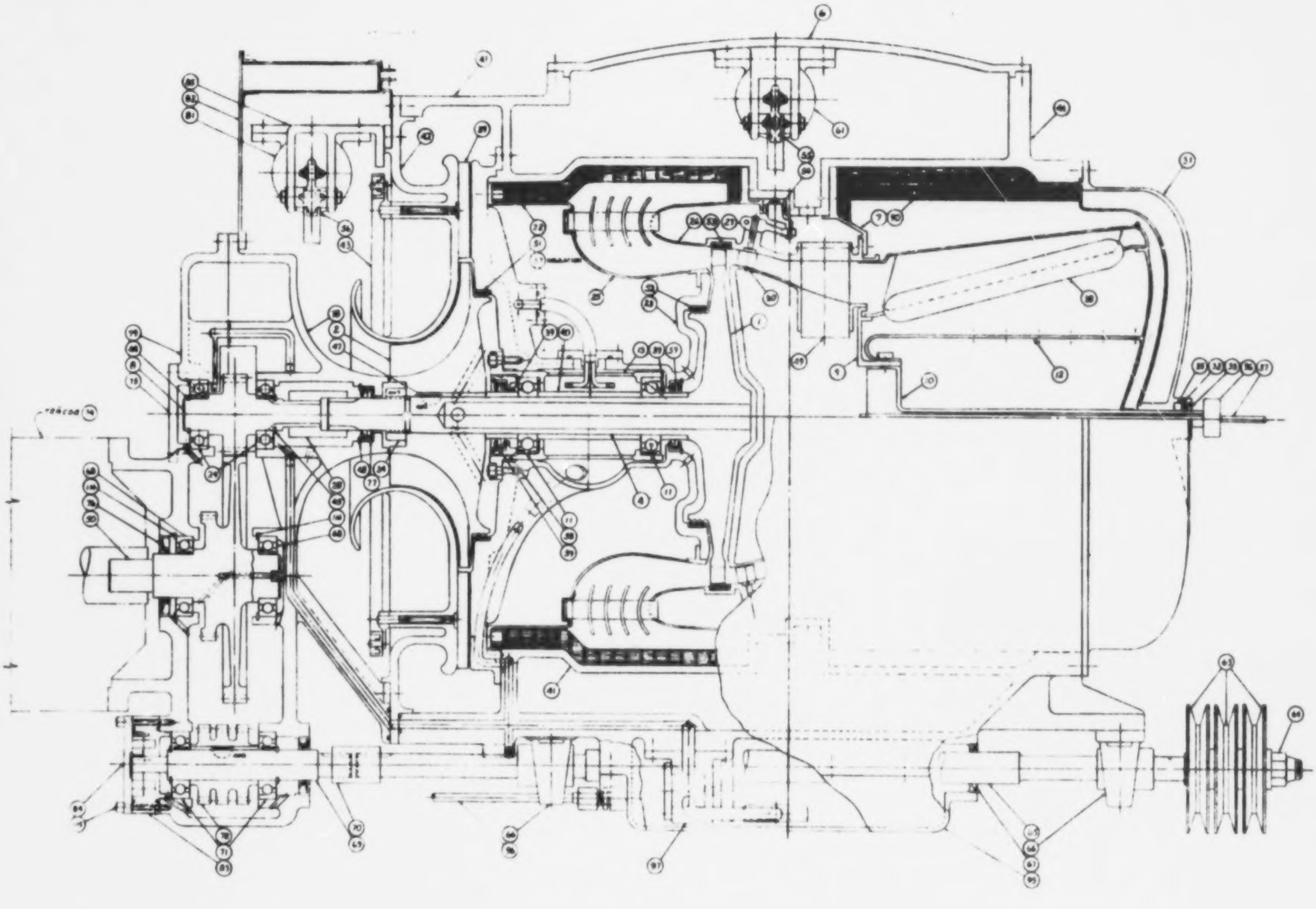

Figure A-2. PD-1 Engine Drawing With Part Numbers. Section AA.

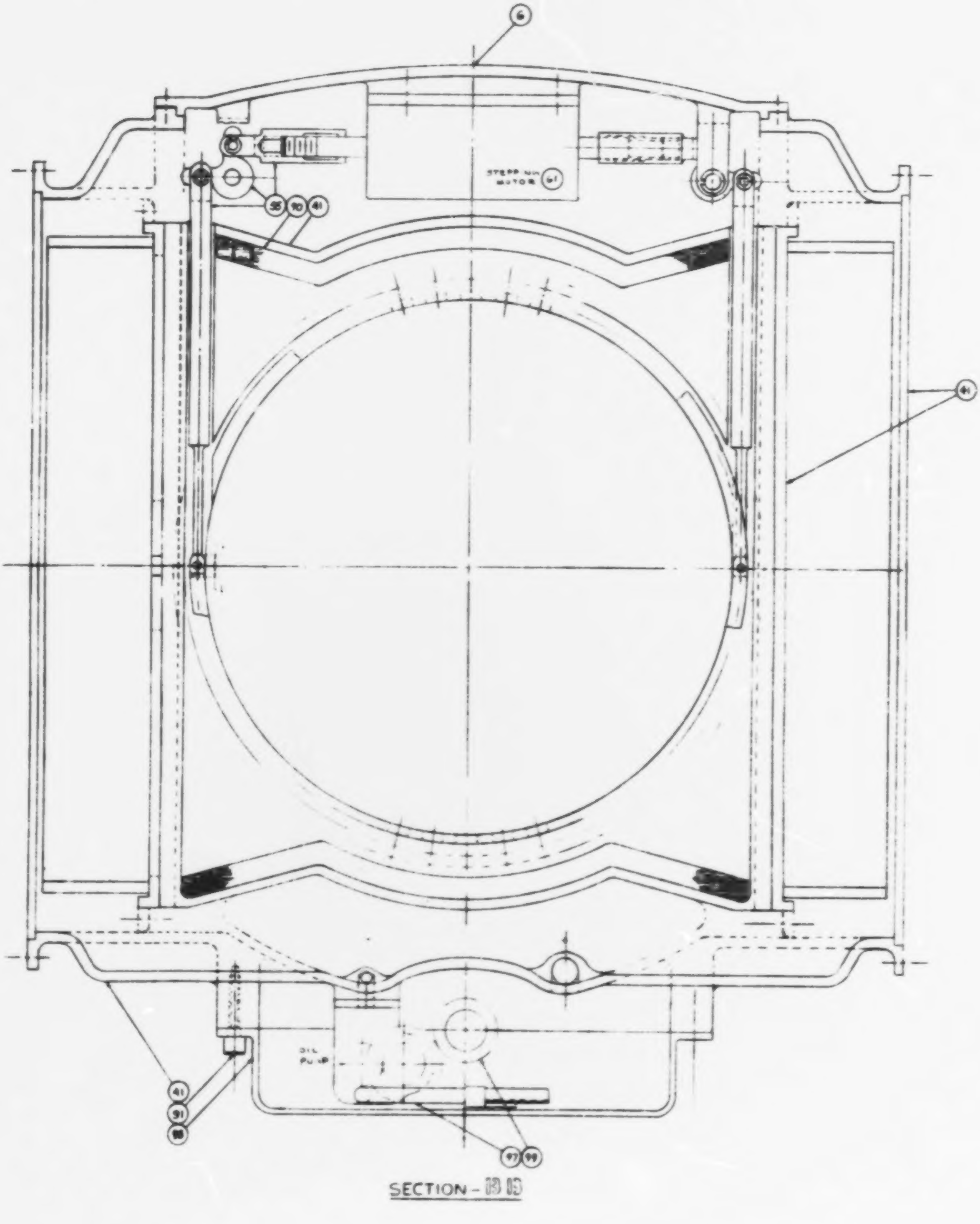
6
STEPPING MOTOR 61
55
90
41
41
41
91
98
OIL PUMP
97
99
SECTION - BB

Figure A-3. PD-1 Engine Drawing With Part Numbers. Section BB.

Table A-6

Single Shaft Engine Parts Breakdown, PD-1

Item	Table No.	Weight (Lbs.)	Cost ($)
Complete Parts List	A-7	483.72	503.22
Castings	A-8	327.03	269.19
Forgings	A-9	19.70	5.91
Bar, Tube, & Extrusions	A-10	45.46	34.03
Sheet & Plate	A-11	20.80	13.66
Purchased Parts	A-12	69.06	153.61
Miscellaneous	A-13	1.67	26.82
Hastelloy X and Inconel 713LC	A-14	22.93	153.23
Steel	A-15	44.92	13.54
Nodular Iron	A-16	232.90	52.79
304 Stainless Steel	A-17	39.09	60.11
Aluminum	A-18	54.79	48.34
Precision Investment Castings	A-19	33.90	171.60

Table A-7

Complete Parts List

AUTOMOTIVE GAS TURBINE SELECTION STUDY
PARTS LIST
PD-1 ENGINE (SINGLE SHAFT)

PART	NAME	DESCRIPTION	MATERIAL	TYPE	COST/LB	WEIGHT	TOTAL WEIGHT	NUMBER OF PARTS	COST PER PART	TOTAL COST
001	TURBINE	-	INC 713LC	PIC	5.50	7.500	7.50	1	41.25	41.25
002	COMPRESSOR	-	C355-T61 AL	PIC	9.50	2.500	2.50	1	23.75	23.75
003	REGENERATOR	-	CERCOR	PUR	0.00	10.000	20.00	2	30.00	60.00
004	SHAFT	MAIN	4340 STEEL	BAR	0.20	1.400	1.40	1	0.28	0.28
005	COVER PLATE	REGENERATOR	NOD. IRON	CST	0.25	17.000	34.00	2	4.25	8.50
006	COVER PLATE	TOP	NOD. IRON	CST	0.30	10.000	10.00	1	3.00	3.00
007	FLANGE	TURBINE INLET REGENERATOR	304 SST	CST	2.00	3.000	3.00	1	6.00	6.00
008	GEAR	MAIN DRIVE	8620 STEEL	FOR	0.30	1.999	2.00	1	0.60	0.60
009	FLANGE	AFT INNER COMBUST SUPPORT	304 SST	CST	2.00	2.170	2.17	1	4.34	4.34
010A	VALVE	COMBUSTOR AIR	304 SST	TUB	0.70	0.432	0.43	1	0.30	0.30
010B	VALVE	COMBUSTOR AIR	304 SST	SHT	0.62	0.432	0.43	1	0.27	0.27
011	BEARING	MAIN SHAFT	-	PUR	0.00	0.522	1.04	2	3.25	6.50
012	PLATE	COMB FWD END-SUP PLENUM	HAST. X	SHT	3.00	1.900	1.90	1	5.70	5.70
013	SUPPORT	MAIN BEARING	NOD. IRON	CST	0.30	1.300	1.30	1	0.39	0.39
014	SHAFT	REGENERATOR	304 SST	BAR	0.62	0.626	1.25	2	0.39	0.78
015	SPACER	REGENERATOR SHAFT	304 SST	EXT	0.70	0.529	1.06	2	0.37	0.74
016	HOUSING	REGENERATOR SHAFT	304 SST	CST	2.00	1.030	2.06	2	2.06	4.12
017	SEAL	REGENERATOR SHAFT	-	PUR	0.00	0.031	0.06	2	0.10	0.20
018	SPROCKET	REGENERATOR DRIVE	8620 STEEL	SHT	0.30	2.202	4.40	2	0.66	1.32
019	BUSHINGS	GRAPHITE REGEN SHAFT	GRAPHITE	TUB	0.80	0.016	0.03	2	0.01	0.03
020	HEX NUT	REGENERATOR SHAFT	304 SST	BAR	0.62	0.035	0.07	2	0.02	0.04
021	SEAL	REGENERATOR SHAFT	-	PUR	0.00	0.012	0.02	2	0.10	0.20
022	INSUL. SHELL	TURB EXIT&REGEN(IN 25)	304 SST	SHT	0.00	0.000	0.00	1	0.00	0.00
023	HOUSING	MAIN BEARING	NOD. IRON	CST	0.25	26.000	26.00	1	6.50	6.50
024	BEARING	MAIN DRIVE GEAR	-	PUR	0.00	0.339	0.68	2	2.80	5.60
025	SHELL	TURBINE EXHAUST INNER	304 SST	SHT	0.62	2.300	2.30	1	1.43	1.43
026A	SHELL	TURB INLET EXHAUST OUTER	304 SST	PIC	2.00	8.250	8.25	1	16.50	16.50
026B	SHELL	TURB INLET EXHAUST OUTER	HAST. X	PIC	6.00	4.500	4.50	1	27.00	27.00
026C	SHELL	TURB INLET EXHAUST OUTER	304 SST	SHT	0.62	3.250	3.25	1	2.02	2.02
027	LINK	NOZZLE VANE ACTUATOR	304 SST	PIC	2.00	0.084	6.05	72	0.17	12.10
028	COUPLING	MAIN SHAFT DRIVE GEAR	4140 STEEL	EXT	0.20	0.444	0.44	1	0.09	0.09
029	VANES	COMPRESSOR DIFFUSER	142 AL	DCS	0.45	0.193	5.98	31	0.09	2.69
030	VANES	TURBINE STATOR	INC 713LC	PIC	10.00	0.051	5.10	100	0.51	51.00

Table A-7 (Cont'd.)

AUTOMOTIVE GAS TURBINE SELECTION STUDY
PARTS LIST
PD-1 ENGINE (SINGLE SHAFT)

PART	NAME	DESCRIPTION	MATERIAL	TYPE	COST/LB	WEIGHT	TOTAL WEIGHT	NUMBER OF PARTS	COST PER PART	TOTAL COST
031	COVER PLATE	FRONT	NOD. IRON	CST	0.25	12.000	12.00	1	3.00	3.00
032	SNAP RING	COMBUSTOR AIR VALVE	-	PUR	0.00	0.014	0.01	1	0.04	0.04
033	SEAL	OUTER COMBUSTOR AIR VALVE	-	PUR	0.00	0.015	0.02	1	0.10	0.10
034	NUT	COMPRESSOR	4340 STEEL	TUB	0.20	0.216	0.22	1	0.04	0.04
035	SEAL	INNER COMB. AIR VALVE	-	PUR	0.00	0.042	0.04	1	0.10	0.10
036	LINK	COMBUSTOR AIR VALVE	4140 STEEL	BAR	0.20	0.260	0.26	1	0.05	0.05
037	TUBE	FUEL SUPPLY	1010 STEEL	TUB	0.18	0.019	0.02	1	0.00	0.00
038	FRIT	COMBUSTOR AND SUPPORTS	-	PUR	0.00	7.000	7.00	1	19.00	19.00
039	SEAL RING	MAIN SHAFT	4340 STEEL	EXT	0.20	0.195	0.39	2	0.04	0.08
040	SPACER	MAIN SHAFT BEARINGS	4340 STEEL	EXT	0.20	0.628	0.63	1	0.13	0.13
041	SUP FRAME	CENTER	NOD. IRON	CST	0.20	140.000	140.00	1	28.00	28.00
042	SCROLL	COMPRESSOR OUTER	333 AL	DCS	0.55	10.600	10.60	1	5.83	5.83
043	ACTUATOR RING	DIFFUSER VANE	333 AL	DCS	0.45	1.425	1.42	1	0.64	0.64
044	SNAP RING	REAR MAIN DRIVE GEAR	4340 STEEL	PUR	0.00	0.013	0.01	1	0.04	0.04
045	SNAP RING	FRONT MAIN DRIVE GEAR	4340 STEEL	PUR	0.00	0.019	0.02	1	0.04	0.04
046	BEARING	AUX. SHAFT DRIVE GEAR	-	PUR	0.00	0.339	0.68	2	2.80	5.60
047	WOODRUFF KEY	SHAFT	4140 STEEL	PUR	0.00	0.011	0.01	1	0.03	0.03
048	SNAP RING	MAIN SHAFT OIL SEAL	4340 STEEL	PUR	0.00	0.006	0.01	2	0.00	0.00
049	STRUT	-	HAST. X	BAR	3.00	0.475	3.80	8	1.42	11.40
050	DRV GEAR ASY	TRANSMISSION	8620 STEEL	FOR	0.30	7.200	7.20	1	2.16	2.16
051	TIP SEAL	COMPRESSOR LABYRINTH	1020 STEEL	HNC	19.30	0.050	0.05	1	0.97	0.97
052	TIP SEAL	TURBINE LABYRINTH	304 SST	HNC	21.00	0.050	0.05	1	1.05	1.05
053	TIP SEAL	TURBINE	HAST. X	HNC	134.00	0.126	0.13	1	16.88	16.88
054	ACT RING	TURBINE NOZZLE	304 SST	CST	2.00	3.100	3.10	1	6.20	6.20
055A	LINKAGE ASSY	TURB NOZZLE ACT RING	304 SST	BAR	0.62	0.320	0.64	2	0.20	0.40
055B	LINKAGE ASSY	TURB NOZZLE ACT RING	304 SST	BAR	0.62	0.520	1.04	2	0.32	0.64
055C	LINKAGE ASSY	TURB NOZZLE ACT RING	304 SST	CST	2.00	0.220	0.44	2	0.44	0.88
055D	LINKAGE ASSY	TURB NOZZLE ACT RING	304 SST	PUR	0.00	0.000	0.00	6	0.01	0.06
055E	LINKAGE ASSY	TURB NOZZLE ACT RING	304 SST	PUR	0.00	0.000	0.00	2	0.04	0.08
056A	LINKAGE ASSY	COMP DIFF VANE ACT RING	4340 STEEL	BAR	0.20	0.640	1.28	2	0.13	0.26
056B	LINKAGE ASSY	COMP DIFF VANE ACT RING	4340 STEEL	BAR	0.20	0.300	0.60	2	0.06	0.12
056C	LINKAGE ASSY	COMP DIFF VANE ACT RING	4340 STEEL	CST	0.40	0.220	0.44	2	0.09	0.18
056D	LINKAGE ASSY	COMP DIFF VANE ACT RING	4340 STEEL	PUR	0.00	0.000	0.00	6	0.01	0.06

Table A-7 (Cont'd.)

AUTOMOTIVE GAS TURBINE SELECTION STUDY
PARTS LIST
PD-1 ENGINE (SINGLE SHAFT)

PART	NAME	DESCRIPTION	MATERIAL	TYPE	COST/LB	WEIGHT	TOTAL WEIGHT	NUMBER OF PARTS	COST PER PART	TOTAL COST
056E	LINKAGE ASSY	COMP DIFF VANE ACT RING	4340 STEEL	PUR	0.00	0.000	0.00	12	0.04	0.48
057	INSULATION	MAIN BEARING HOUSING	-	PUR	0.20	2.800	2.80	1	0.56	0.56
058	SEAL HOUSING	COMPRESSOR BEARING	NOD. IRON	CST	0.30	0.372	0.37	1	0.11	0.11
059	SEAL	OIL MAIN SHAFT	-	PUR	0.00	0.091	0.18	2	0.10	0.20
060	DRV GEAR ASSY	ACCESSORY	8620 STEEL	FOR	0.30	5.100	5.10	1	1.53	1.53
061	MOTOR	TURB. NOZZLE VANE ACT.	-	PUR	0.00	3.000	3.00	1	2.00	2.00
062	VALVE	FUEL SUPPLY	-	PUR	0.00	0.333	0.33	1	0.50	0.50
063	SHEAVE	V BELT	NOD. IRON	PUR	0.00	0.400	1.20	3	0.26	0.78
064	NUT	ACCESSORY DRIVE SHAFT	4140 STEEL	BAR	0.20	0.052	0.05	1	0.01	0.01
065	SHAFT	AUXILIARY DRIVE	4140STEEL	BAR	0.20	4.988	4.99	1	1.00	1.00
066	BEARING	AUXILIARY SHAFT	-	PUR	0.00	1.444	2.89	2	2.80	5.60
067	SEAL	AUXILIARY SHAFT	-	PUR	0.00	0.035	0.07	2	0.10	0.20
068	BEARING	TRANSMISSION DRIVE GEAR	-	PUR	0.00	0.745	1.49	2	3.80	7.60
069	SHAFT	CHAINBELT	4140 STEEL	BAR	0.20	1.138	1.14	1	0.23	0.23
070	COUPLING	CHAINBELT AUX DRIVE SHAFT	4140 STEEL	TUB	0.20	0.253	0.25	1	0.05	0.05
071	BEARING	CHAIN BELT SHAFT	-	PUR	0.00	0.544	1.09	2	2.80	5.60
072	SPROCKET	CHAINBELT DRV	8620 STEEL	FOR	0.30	0.437	1.31	3	0.13	0.39
073	SNAP RING	REG.DRIVE SHAFT	4340 STEEL	PUR	0.00	0.008	0.02	2	0.04	0.08
075	COVER PLATE	MAIN SHAFT AFT BEARING	43 AL	DCS	0.45	0.234	0.23	1	0.11	0.11
076	SEAL	TRANS. DRIVE GEAR	-	PUR	0.00	0.094	0.09	1	0.10	0.10
077	SEAL	OIL DRIVE GEAR FWD.	-	PUR	0.00	0.079	0.08	1	0.10	0.10
078	SCROLL	COMPRESSOR INNER	43 AL	DCS	0.45	15.000	15.00	1	6.75	6.75
079	COVER PLATE	AFT	43 AL	DCS	0.45	8.100	8.10	1	3.65	3.65
081	MOTOR	DIFFUSER VANE ACT.	-	PUR	0.00	3.000	3.00	1	2.00	2.00
082	AIR FILTER	ASSEMBLY	1010 STEEL	SHT	0.15	2.017	2.02	1	0.30	0.30
083	OIL PUMP	SCAVENGE	NOD. IRON	PUR	0.00	1.580	1.58	1	0.55	0.55
084	COVER	SCAVENGE OIL PUMP	NOD. IRON	PUR	0.00	0.485	0.48	1	0.17	0.17
085	BRACKET	DIFFUSER VANE ACT MOTOR	43 AL	DCS	0.45	6.100	6.10	1	2.75	2.75
086	COVER PLATE	REGENERATOR DRIVE	1010 STEEL	SHT	0.15	0.600	1.20	2	0.09	0.18
087	HOUSING	REGENERATOR DRIVE	1010 STEEL	SHT	0.15	0.900	1.80	2	0.13	0.27
088	SNAP RING	REG.DRIVE SHAFT	4340 STEEL	PUR	0.00	0.020	0.04	2	0.04	0.08
089	CHAIN	REG.DRIVE GEAR	-	PUR	0.00	1.019	2.04	2	2.50	5.00
090	INSULATION	REGENERATOR	-	PUR	0.20	10.000	10.00	1	2.00	2.00

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Table A-7 (Cont'd.)

AUTOMOTIVE GAS TURBINE SELECTION STUDY
PARTS LIST
PD-1 ENGINE (SINGLE SHAFT)

PART	NAME	DESCRIPTION	MATERIAL	TYPE	COST/LB	WEIGHT	TOTAL WEIGHT	NUMBER OF PARTS	COST PER PART	TOTAL COST
091	BOLT	OIL SUMP	4140 STEEL	PUR	0.00	0.000	0.00	2	0.01	0.02
092	SEAL PLATES	REGENERATOR	304 SS	EXT	0.70	5.123	10.25	2	3.59	7.17
093A	SEAL	REGENERATOR	NIO+CA2F	PLA	5.50	0.290	0.58	2	1.59	3.19
093B	SEAL	REGENERATOR	NIO+CA2F	ARC	5.50	0.430	0.86	2	2.37	4.73
093C	SEAL	REGENERATOR	304 SS	EXT	0.70	1.280	2.56	2	0.90	1.79
093D	SEAL	REGENERATOR	304 SS	EXT	0.70	0.860	1.72	2	0.60	1.20
094	PLATE	REGENERATOR	304 SS	EXT	0.70	4.600	9.20	2	3.22	6.44
095	OIL SUMP	-	NOD. IRON	CST	0.30	5.500	5.50	1	1.65	1.65
096	TUBING SET	OIL SUMP SCAVENGE PUMP	MILD STEEL	PUR	0.00	2.814	2.81	1	1.05	1.05
097	OIL PUMP	MAIN(PLUS FILTER)	-	PUR	0.00	3.569	3.57	1	2.15	2.15
098	CAN	SPEED REDUCER FRONT HALF	43 AL	DCS	0.45	1.252	2.50	2	0.56	1.13
099	DRIVE GEAR	MAIN OIL PUMP	8620 STEEL	FOR	0.30	0.453	0.45	1	0.14	0.14
100	CASE	SPEED REDUCER REAR HALF	33 AL	DCS	0.45	1.174	2.35	2	0.53	1.06
101	WORM GEAR	-	8620 STEEL	FOR	0.30	0.316	0.63	2	0.09	0.19
102	WORM	-	8620 STEEL	FOR	0.30	0.481	0.96	2	0.14	0.29
103	CHAINBELT	SPEED REDUCER	-	PUR	0.00	0.664	1.33	2	2.50	5.00
104	CHAINBELT	-	-	PUR	0.00	0.411	0.41	1	2.00	2.00
105	THRUST BEAR	SPEED REDUCER	BRONZE	TUB	0.60	0.126	0.50	4	0.08	0.30
106	SLEEVE BEAR	SPEED REDUCER	BRONZE	TUB	0.60	0.065	0.52	8	0.04	0.31
107	SEAL	OIL SPEED REDUCER	-	PUR	0.00	0.050	0.20	4	0.10	0.40
108	SCREW	FILISTER HEAD	-	PUR	0.00	0.007	0.10	14	0.01	0.14
109	COUPLING	SPLINED	4340 STEEL	TUB	0.20	0.360	0.72	2	0.07	0.14
110	END CAP	-	NOD. IRON	CST	0.30	0.230	0.46	2	0.07	0.14
111	SPROCKET	REGENERATOR DRIVE	8620 STEEL	FOR	0.30	1.020	2.04	2	0.31	0.61
112	BEARING	REG. DRIVE SPROCKET	-	PUR	0.00	0.122	0.49	4	2.80	11.20
113	SEAL	OIL REG. DRIVE SPR.	-	PUR	0.00	0.050	0.10	2	0.10	0.20
114	WOODRUFF KEY	REG. DRIVE SHAFT	4340 STEEL	PUR	0.00	0.000	0.00	2	0.03	0.06
115	SCREW	AUX. OIL PUMP COVER	-	PUR	0.00	0.000	0.00	6	0.01	0.06
116	SNAP RING	TRANS.DRIVE BEARING	-	PUR	0.00	0.028	0.06	2	0.04	0.08
117	SHELL	TURBINE NOZZLE INNER	304 SST	SHT	0.62	3.500	3.50	1	2.17	2.17

TOTAL WEIGHT = 483.72 LBS. TOTAL COST = $503.22 NUMBER OF PARTS = 445

Table A-8

Castings

AUTOMOTIVE GAS TURBINE SELECTION STUDY
PARTS LIST
PD-1 ENGINE (SINGLE SHAFT)

PART	NAME	DESCRIPTION	MATERIAL	TYPE	COST/LB	WEIGHT	TOTAL WEIGHT	NUMBER OF PARTS	COST PER PART	TOTAL COST
001	TURBINE	-	INC 713LC	PIC	5.50	7.500	7.50	1	41.25	41.25
002	COMPRESSOR	-	C355-T61 AL	PIC	9.50	2.500	2.50	1	23.75	23.75
005	COVER PLATE	REGENERATOR	NOD. IRON	CST	0.25	17.000	34.00	2	4.25	8.50
006	COVER PLATE	TOP	NOD. IRON	CST	0.30	10.000	10.00	1	3.00	3.00
007	FLANGE	TURBINE INLET REGENERATOR	304 SST	CST	2.00	3.000	3.00	1	6.00	6.00
009	FLANGE	AFT INNER COMBUST SUPPORT	304 SST	CST	2.00	2.170	2.17	1	4.34	4.34
013	SUPPORT	MAIN BEARING	NOD. IRON	CST	0.30	1.300	1.30	1	0.39	0.39
016	HOUSING	REGENERATOR SHAFT	304 SST	CST	2.00	1.030	2.06	2	2.06	4.12
023	HOUSING	MAIN BEARING	NOD. IRON	CST	0.25	26.000	26.00	1	6.50	6.50
026A	SHELL	TURB INLET EXHAUST OUTER	304 SST	PIC	2.00	8.250	8.25	1	16.50	16.50
026B	SHELL	TURB INLET EXHAUST OUTER	HAST. X	PIC	6.00	4.500	4.50	1	27.00	27.00
027	LINK	NOZZLE VANE ACTUATOR	304 SST	PIC	2.00	0.084	6.05	72	0.17	12.10
029	VANES	COMPRESSOR DIFFUSER	142 AL	DCS	0.45	0.193	5.98	31	0.09	2.69
030	VANES	TURBINE STATOR	INC 713LC	PIC	10.00	0.051	5.10	100	0.51	51.00
031	COVER PLATE	FRONT	NOD. IRON	CST	0.25	12.000	12.00	1	3.00	3.00
041	SUP FRAME	CENTER	NOD. IRON	CST	0.20	140.000	140.00	1	28.00	28.00
042	SCROLL	COMPRESSOR OUTER	333 AL	DCS	0.55	10.600	10.60	1	5.83	5.83
043	ACTUATOR RING	DIFFUSER VANE	333 AL	DCS	0.45	1.425	1.42	1	0.64	0.64
054	ACT RING	TURBINE NOZZLE	304 SST	CST	2.00	3.100	3.10	1	6.20	6.20
055C	LINKAGE ASSY	TURB NOZZLE ACT RING	304 SST	CST	2.00	0.220	0.44	2	0.44	0.88
056C	LINKAGE ASSY	COMP DIFF VANE ACT RING	4340 STEEL	CST	0.40	0.220	0.44	2	0.09	0.18
058	SEAL HOUSING	COMPRESSOR BEARING	NOD. IRON	CST	0.30	0.372	0.37	1	0.11	0.11
075	COVER PLATE	MAIN SHAFT AFT BEARING	43 AL	DCS	0.45	0.234	0.23	1	0.11	0.11
078	SCROLL	COMPRESSOR INNER	43 AL	DCS	0.45	15.000	15.00	1	6.75	6.75
079	COVER PLATE	AFT	43 AL	DCS	0.45	8.100	8.10	1	3.65	3.65
085	BRACKET	DIFFUSER VANE ACT MOTOR	43 AL	DCS	0.45	6.100	6.10	1	2.75	2.75
095	OIL SUMP	-	NOD. IRON	CST	0.30	5.500	5.50	1	1.65	1.65
098	CAN	SPEED REDUCER FRONT HALF	43 AL	DCS	0.45	1.252	2.50	2	0.56	1.13
100	CASE	SPEED REDUCER REAR HALF	33 AL	DCS	0.45	1.174	2.35	2	0.53	1.06
110	END CAP	-	NOD. IRON	CST	0.30	0.230	0.46	2	0.07	0.14

TOTAL WEIGHT = 327.03 LBS. TOTAL COST = $269.19 NUMBER OF PARTS = 237

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Table A-9

Forgings

AUTOMOTIVE GAS TURBINE SELECTION STUDY
PARTS LIST
PD-1 ENGINE (SINGLE SHAFT)

PART	NAME	DESCRIPTION	MATERIAL	TYPE	COST/LB	WEIGHT	TOTAL WEIGHT	NUMBER OF PARTS	COST PER PART	TOTAL COST
008	GEAR	MAIN DRIVE	8620 STEEL	FOR	0.30	1.999	2.00	1	0.60	0.60
050	DRV GEAR ASY	TRANSMISSION	8620 STEEL	FOR	0.30	7.200	7.20	1	2.16	2.16
060	DRV GEAR ASSY	ACCESSORY	8620 STEEL	FOR	0.30	5.100	5.10	1	1.53	1.53
072	SPROCKET	CHAINBELT DRV	8620 STEEL	FOR	0.30	0.437	1.31	3	0.13	0.39
099	DRIVE GEAR	MAIN OIL PUMP	8620 STEEL	FOR	0.30	0.453	0.45	1	0.14	0.14
101	WORM GEAR	-	8620 STEEL	FOR	0.30	0.316	0.63	2	0.09	0.19
102	WORM	-	8620 STEEL	FOR	0.30	0.481	0.96	2	0.14	0.29
111	SPROCKET	REGENERATOR DRIVE	8620 STEEL	FOR	0.30	1.020	2.04	2	0.31	0.61

TOTAL WEIGHT = 19.70 LBS. TOTAL COST = $ 5.91 NUMBER OF PARTS = 13

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Table A-10

Bar, Tube, & Extrusions

AUTOMOTIVE GAS TURBINE SELECTION STUDY
PARTS LIST
PD-1 ENGINE (SINGLE SHAFT)

PART	NAME	DESCRIPTION	MATERIAL	TYPE	COST/LB	WEIGHT	TOTAL WEIGHT	NUMBER OF PARTS	COST PER PART	TOTAL COST
004	SHAFT	MAIN	4340 STEEL	BAR	0.20	1.400	1.40	1	0.28	0.28
010A	VALVE	COMBUSTOR AIR	304 SST	TUB	0.70	0.432	0.43	1	0.30	0.30
014	SHAFT	REGENERATOR	304 SST	BAR	0.62	0.626	1.25	2	0.39	0.78
015	SPACER	REGENERATOR SHAFT	304 SST	EXT	0.70	0.529	1.06	2	0.37	0.74
019	BUSHINGS	GRAPHITE REGEN SHAFT	GRAPHITE	TUB	0.80	0.016	0.03	2	0.01	0.03
020	HEX NUT	REGENERATOR SHAFT	304 SST	BAR	0.62	0.035	0.07	2	0.02	0.04
028	COUPLING	MAIN SHAFT DRIVE GEAR	4140 STEEL	EXT	0.20	0.444	0.44	1	0.09	0.09
034	NUT	COMPRESSOR	4340 STEEL	TUB	0.20	0.216	0.22	1	0.04	0.04
036	LINK	COMBUSTOR AIR VALVE	4140 STEEL	BAR	0.20	0.260	0.26	1	0.05	0.05
037	TUBE	FUEL SUPPLY	1010 STEEL	TUB	0.18	0.019	0.02	1	0.00	0.00
039	SEAL RING	MAIN SHAFT	4340 STEEL	EXT	0.20	0.195	0.39	2	0.04	0.08
040	SPACER	MAIN SHAFT BEARINGS	4340 STEEL	EXT	0.20	0.628	0.63	1	0.13	0.13
049	STRUT	-	HAST. X	BAR	3.00	0.475	3.80	8	1.42	11.40
055A	LINKAGE ASSY	TURB NOZZLE ACT RING	304 SST	BAR	0.62	0.320	0.64	2	0.20	0.40
055B	LINKAGE ASSY	TURB NOZZLE ACT RING	304 SST	BAR	0.62	0.520	1.04	2	0.32	0.64
056A	LINKAGE ASSY	COMP DIFF VANE ACT RING	4340 STEEL	BAR	0.20	0.640	1.28	2	0.13	0.26
056B	LINKAGE ASSY	COMP DIFF VANE ACT RING	4340 STEEL	BAR	0.20	0.300	0.60	2	0.06	0.12
064	NUT	ACCESSORY DRIVE SHAFT	4140 STEEL	BAR	0.20	0.052	0.05	1	0.01	0.01
065	SHAFT	AUXILIARY DRIVE	4140STEEL	BAR	0.20	4.988	4.99	1	1.00	1.00
069	SHAFT	CHAINBELT	4140 STEEL	BAR	0.20	1.138	1.14	1	0.23	0.23
070	COUPLING	CHAINBELT AUX DRIVE SHAFT	4140 STEEL	TUB	0.20	0.253	0.25	1	0.05	0.05
092	SEAL PLATES	REGENERATOR	304 SS	EXT	0.70	5.123	10.25	2	3.59	7.17
093C	SEAL	REGENERATOR	304 SS	EXT	0.70	1.280	2.56	2	0.90	1.79
093D	SEAL	REGENERATOR	304 SS	EXT	0.70	0.860	1.72	2	0.60	1.20
094	PLATE	REGENERATOR	304 SS	EXT	0.70	4.600	9.20	2	3.22	6.44
105	THRUST BEAR	SPEED REDUCER	BRONZE	TUB	0.60	0.126	0.50	4	0.08	0.30
106	SLEEVE BEAR	SPEED REDUCER	BRONZE	TUB	0.60	0.065	0.52	8	0.04	0.31
109	COUPLING	SPLINED	4340 STEEL	TUB	0.20	0.360	0.72	2	0.07	0.14

TOTAL WEIGHT = 45.46 LBS. TOTAL COST = $ 34.03 NUMBER OF PARTS = 59

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Table A-11

Sheet & Plate

AUTOMOTIVE GAS TURBINE SELECTION STUDY
PARTS LIST
PD-1 ENGINE (SINGLE SHAFT)

PART	NAME	DESCRIPTION	MATERIAL	TYPE	COST/LB	WEIGHT	TOTAL WEIGHT	NUMBER OF PARTS	COST PER PART	TOTAL COST
010B	VALVE	COMBUSTOR AIR	304 SST	SHT	0.62	0.432	0.43	1	0.27	0.27
012	PLATE	COMB FWD END-SUP PLENUM	HAST. X	SHT	3.00	1.900	1.90	1	5.70	5.70
018	SPROCKET	REGENERATOR DRIVE	8620 STEEL	SHT	0.30	2.202	4.40	2	0.66	1.32
022	INSUL. SHELL	TURB EXIT&REGEN(IN 25)	304 SST	SHT	0.00	0.000	0.00	1	0.00	0.00
025	SHELL	TURBINE EXHAUST INNER	304 SST	SHT	0.62	2.300	2.30	1	1.43	1.43
026C	SHELL	TURB INLET EXHAUST OUTER	304 SST	SHT	0.62	3.250	3.25	1	2.02	2.02
082	AIR FILTER	ASSEMBLY	1010 STEEL	SHT	0.15	2.017	2.02	1	0.30	0.30
086	COVER PLATE	REGENERATOR DRIVE	1010 STEEL	SHT,	0.15	0.600	1.20	2	0.09	0.18
087	HOUSING	REGENERATOR DRIVE	1010 STEEL	SHT	0.15	0.900	1.80	2	0.13	0.27
117	SHELL	TURBINE NOZZLE INNER	304 SST	SHT	0.62	3.500	3.50	1	2.17	2.17

TOTAL WEIGHT = 20.80 LBS. TOTAL COST = $ 13.66 NUMBER OF PARTS = 13

Table A-12

Purchased Parts

AUTOMOTIVE GAS TURBINE SELECTION STUDY
PARTS LIST
PD-1 ENGINE (SINGLE SHAFT)

PART	NAME	DESCRIPTION	MATERIAL	TYPE	COST/LB	WEIGHT	TOTAL WEIGHT	NUMBER OF PARTS	COST PER PART	TOTAL COST
003	REGENERATOR	-	CERCOR	PUR	0.00	10.000	20.00	2	30.00	60.00
011	BEARING	MAIN SHAFT	-	PUR	0.00	0.522	1.04	2	3.25	6.50
017	SEAL	REGENERATOR SHAFT	-	PUR	0.00	0.031	0.06	2	0.10	0.20
021	SEAL	REGENERATOR SHAFT	-	PUR	0.00	0.012	0.02	2	0.10	0.20
024	BEARING	MAIN DRIVE GEAR	-	PUR	0.00	0.339	0.68	2	2.80	5.60
032	SNAP RING	COMBUSTOR AIR VALVE	-	PUR	0.00	0.014	0.01	1	0.04	0.04
033	SEAL	OUTER COMBUSTOR AIR VALVE	-	PUR	0.00	0.015	0.02	1	0.10	0.10
035	SEAL	INNER COMB. AIR VALVE	-	PUR	0.00	0.042	0.04	1	0.10	0.10
038	FRIT	COMBUSTOR AND SUPPORTS	-	PUR	0.00	7.000	7.00	1	19.00	19.00
044	SNAP RING	REAR MAIN DRIVE GEAR	4340 STEEL	PUR	0.00	0.013	0.01	1	0.04	0.04
045	SNAP RING	FRONT MAIN DRIVE GEAR	4340 STEEL	PUR	0.00	0.019	0.02	1	0.04	0.04
046	BEARING	AUX. SHAFT DRIVE GEAR	-	PUR	0.00	0.339	0.68	2	2.80	5.60
047	WOODRUFF KEY	SHAFT	4140 STEEL	PUR	0.00	0.011	0.01	1	0.03	0.03
048	SNAP RING	MAIN SHAFT OIL SEAL	4340 STEEL	PUR	0.00	0.006	0.01	2	0.00	0.00
055D	LINKAGE ASSY	TURB NOZZLE ACT RING	304 SST	PUR	0.00	0.000	0.00	6	0.01	0.06
055E	LINKAGE ASSY	TURB NOZZLE ACT RING	304 SST	PUR	0.00	0.000	0.00	2	0.04	0.08
056D	LINKAGE ASSY	COMP DIFF VANE ACT RING	4340 STEEL	PUR	0.00	0.000	0.00	6	0.01	0.06
056E	LINKAGE ASSY	COMP DIFF VANE ACT RING	4340 STEEL	PUR	0.00	0.000	0.00	12	0.04	0.48
057	INSULATION	MAIN BEARING HOUSING	-	PUR	0.20	2.800	2.80	1	0.56	0.56
059	SEAL	OIL MAIN SHAFT	-	PUR	0.00	0.091	0.18	2	0.10	0.20
061	MOTOR	TURB. NOZZLE VANE ACT.	-	PUR	0.00	3.000	3.00	1	2.00	2.00
062	VALVE	FUEL SUPPLY	-	PUR	0.00	0.333	0.33	1	0.50	0.50
063	SHEAVE	V BELT	NOD. IRON	PUR	0.00	0.400	1.20	3	0.26	0.78
066	BEARING	AUXILIARY SHAFT	-	PUR	0.00	1.444	2.89	2	2.80	5.60
067	SEAL	AUXILIARY SHAFT	-	PUR	0.00	0.035	0.07	2	0.10	0.20
068	BEARING	TRANSMISSION DRIVE GEAR	-	PUR	0.00	0.745	1.49	2	3.80	7.60
071	BEARING	CHAIN BELT SHAFT	-	PUR	0.00	0.544	1.09	2	2.80	5.60
073	SNAP RING	REG.DRIVE SHAFT	4340 STEEL	PUR	0.00	0.008	0.02	2	0.04	0.08
074	SEAL	TRANS. DRIVE GEAR	-	PUR	0.00	0.094	0.09	1	0.10	0.10
077	SEAL	OIL DRIVE GEAR FWD.	-	PUR	0.00	0.079	0.08	1	0.10	0.10
081	MOTOR	DIFFUSER VANE ACT.	-	PUR	0.00	3.000	3.00	1	2.00	2.00
083	OIL PUMP	SCAVENGE	NOD. IRON	PUR	0.00	1.580	1.58	1	0.55	0.55
084	COVER	SCAVENGE OIL PUMP	NOD. IRON	PUR	0.00	0.485	0.48	1	0.17	0.17

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Table A-12 (Cont'd.)

AUTOMOTIVE GAS TURBINE SELECTION STUDY
PARTS LIST
PD-1 ENGINE (SINGLE SHAFT)

PART	NAME	DESCRIPTION	MATERIAL	TYPE	COST/LB	WEIGHT	TOTAL WEIGHT	NUMBER OF PARTS	COST PER PART	TOTAL COST
088	SNAP RING	REG.DRIVE SHAFT	4340 STEEL	PUR	0.00	0.020	0.04	2	0.04	0.08
089	CHAIN	REG.DRIVE GEAR	-	PUR	0.00	1.019	2.04	2	2.50	5.00
090	INSULATION	REGENERATOR	-	PUR	0.20	10.000	10.00	1	2.00	2.00
091	BOLT	OIL SUMP	4140 STEEL	PUR	0.00	0.000	0.00	2	0.01	0.02
096	TUBING SET	OIL SUMP SCAVENGE PUMP	MILD STEEL	PUR	0.00	2.814	2.81	1	1.05	1.05
097	OIL PUMP	MAIN(PLUS FILTER)	-	PUR	0.00	3.569	3.57	1	2.15	2.15
103	CHAINBELT	SPEED REDUCER	-	PUR	0.00	0.664	1.33	2	2.50	5.00
104	CHAINBELT	-	-	PUR	0.00	0.411	0.41	1	2.00	2.00
107	SEAL	OIL SPEED REDUCER	-	PUR	0.00	0.050	0.20	4	0.10	0.40
108	SCREW	FILISTER HEAD	-	PUR	0.00	0.007	0.10	14	0.01	0.14
112	BEARING	REG. DRIVE SPROCKET	-	PUR	0.00	0.122	0.49	4	2.80	11.20
113	SEAL	OIL REG. DRIVE SPR.	-	PUR	0.00	0.050	0.10	2	0.10	0.20
114	WOODRUFF KEY	REG. DRIVE SHAFT	4340 STEEL	PUR	0.00	0.000	0.00	2	0.03	0.06
115	SCREW	AUX. OIL PUMP COVER	-	PUR	0.00	0.000	0.00	6	0.01	0.06
116	SNAP RING	TRANS.DRIVE BEARING	-	PUR	0.00	0.028	0.06	2	0.04	0.08

TOTAL WEIGHT = 69.06 LBS. TOTAL COST = $153.61 NUMBER OF PARTS = 116

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Table A-13

Miscellaneous

AUTOMOTIVE GAS TURBINE SELECTION STUDY
PARTS LIST
PD-1 ENGINE (SINGLE SHAFT)

PART	NAME	DESCRIPTION	MATERIAL	TYPE	COST/LB	WEIGHT	TOTAL WEIGHT	NUMBER OF PARTS	COST PER PART	TOTAL COST
051	TIP SEAL	COMPRESSOR LABYRINTH	1020 STEEL	HNC	19.30	0.050	0.05	1	0.97	0.97
052	TIP SEAL	TURBINE LABYRINTH	304 SST	HNC	21.00	0.050	0.05	1	1.05	1.05
053	TIP SEAL	TURBINE	HAST. X	HNC	134.00	0.126	0.13	1	16.88	16.88
093A	SEAL	REGENERATOR	NIO+CA2F	PLA	5.50	0.290	0.58	2	1.59	3.19
093B	SEAL	REGENERATOR	NIO+CA2F	ARC	5.50	0.430	0.86	2	2.37	4.73

TOTAL WEIGHT = 1.67 LBS. TOTAL COST = $ 26.82 NUMBER OF PARTS = 7

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Table A-14

Hastelloy X and Inconel 713LC

AUTOMOTIVE GAS TURBINE SELECTION STUDY
PARTS LIST
PD-1 ENGINE (SINGLE SHAFT)

PART	NAME	DESCRIPTION	MATERIAL	TYPE	COST/LB	WEIGHT	TOTAL WEIGHT	NUMBER OF PARTS	COST PER PART	TOTAL COST
001	TURBINE	-	INC 713LC	PIC	5.50	7.500	7.50	1	41.25	41.25
012	PLATE	COMB FWD END-SUP PLENUM	HAST. X	SHT	3.00	1.900	1.90	1	5.70	5.70
026B	SHELL	TURB INLET EXHAUST OUTER	HAST. X	PIC	6.00	4.500	4.50	1	27.00	27.00
030	VANES	TURBINE STATOR	INC 713LC	PIC	10.00	0.051	5.10	100	0.51	51.00
049	STRUT	-	HAST. X	BAR	3.00	0.475	3.80	8	1.42	11.40
053	TIP SEAL	TURBINE	HAST. X	HNC	134.00	0.126	0.13	1	16.88	16.88

TOTAL WEIGHT = 22.93 LBS. TOTAL COST = $153.23 NUMBER OF PARTS = 112

Table A-15

Steel

AUTOMOTIVE GAS TURBINE SELECTION STUDY
PARTS LIST
PD-1 ENGINE (SINGLE SHAFT)

PART	NAME	DESCRIPTION	MATERIAL	TYPE	COST/LB	WEIGHT	TOTAL WEIGHT	NUMBER OF PARTS	COST PER PART	TOTAL COST
004	SHAFT	MAIN	4340 STEEL	BAR	0.20	1.400	1.40	1	0.28	0.28
008	GEAR	MAIN DRIVE	8620 STEEL	FOR	0.30	1.999	2.00	1	0.60	0.60
018	SPROCKET	REGENERATOR DRIVE	8620 STEEL	SHT	0.30	2.202	4.40	2	0.66	1.32
028	COUPLING	MAIN SHAFT DRIVE GEAR	4140 STEEL	EXT	0.20	0.444	0.44	1	0.09	0.09
034	NUT	COMPRESSOR	4340 STEEL	TUB	0.20	0.216	0.22	1	0.04	0.04
036	LINK	COMBUSTOR AIR VALVE	4140 STEEL	BAR	0.20	0.260	0.26	1	0.05	0.05
037	TUBE	FUEL SUPPLY	1010 STEEL	TUB	0.18	0.019	0.02	1	0.00	0.00
039	SEAL RING	MAIN SHAFT	4340 STEEL	EXT	0.20	0.195	0.39	2	0.04	0.08
040	SPACER	MAIN SHAFT BEARINGS	4340 STEEL	EXT	0.20	0.628	0.63	1	0.13	0.13
044	SNAP RING	REAR MAIN DRIVE GEAR	4340 STEEL	PUR	0.00	0.013	0.01	1	0.04	0.04
045	SNAP RING	FRONT MAIN DRIVE GEAR	4340 STEEL	PUR	0.00	0.019	0.02	1	0.04	0.04
047	WOODRUFF KEY	SHAFT	4140 STEEL	PUR	0.00	0.011	0.01	1	0.03	0.03
048	SNAP RING	MAIN SHAFT OIL SEAL	4340 STEEL	PUR	0.00	0.006	0.01	2	0.00	0.00
050	DRV GEAR ASY	TRANSMISSION	8620 STEEL	FOR	0.30	7.200	7.20	1	2.16	2.16
051	TIP SEAL	COMPRESSOR LABYRINTH	1020 STEEL	HNC	19.30	0.050	0.05	1	0.97	0.97
056A	LINKAGE ASSY	COMP DIFF VANE ACT RING	4340 STEEL	BAR	0.20	0.640	1.28	2	0.13	0.26
056B	LINKAGE ASSY	COMP DIFF VANE ACT RING	4340 STEEL	BAR	0.20	0.300	0.60	2	0.06	0.12
056C	LINKAGE ASSY	COMP DIFF VANE ACT RING	4340 STEEL	CST	0.40	0.220	0.44	2	0.09	0.18
056D	LINKAGE ASSY	COMP DIFF VANE ACT RING	4340 STEEL	PUR	0.00	0.000	0.00	6	0.01	0.06
056E	LINKAGE ASSY	COMP DIFF VANE ACT RING	4340 STEEL	PUR	0.00	0.000	0.00	12	0.04	0.48
060	DRV GEAR ASSY	ACCESSORY	8620 STEEL	FOR	0.30	5.100	5.10	1	1.53	1.53
064	NUT	ACCESSORY DRIVE SHAFT	4140 STEEL	BAR	0.20	0.052	0.05	1	0.01	0.01
065	SHAFT	AUXILIARY DRIVE	4140STEEL	BAR	0.20	4.988	4.99	1	1.00	1.00
069	SHAFT	CHAINBELT	4140 STEEL	BAR	0.20	1.138	1.14	1	0.23	0.23
070	COUPLING	CHAINBELT AUX DRIVE SHAFT	4140 STEEL	TUB	0.20	0.253	0.25	1	0.05	0.05
072	SPROCKET	CHAINBELT DRV	8620 STEEL	FOR	0.30	0.437	1.31	3	0.13	0.39
073	SNAP RING	REG.DRIVE SHAFT	4340 STEEL	PUR	0.00	0.008	0.02	2	0.04	0.08
082	AIR FILTER	ASSEMBLY	1010 STEEL	SHT	0.15	2.017	2.02	1	0.30	0.30
086	COVER PLATE	REGENERATOR DRIVE	1010 STEEL	SHT	0.15	0.600	1.20	2	0.09	0.18
087	HOUSING	REGENERATOR DRIVE	1010 STEEL	SHT	0.15	0.900	1.80	2	0.13	0.27
088	SNAP RING	REG.DRIVE SHAFT	4340 STEEL	PUR	0.00	0.020	0.04	2	0.04	0.08
091	BOLT	OIL SUMP	4140 STEEL	PUR	0.00	0.000	0.00	2	0.01	0.02
096	TUBING SET	OIL SUMP SCAVENGE PUMP	MILD STEEL	PUR	0.00	2.814	2.81	1	1.05	1.05

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Table A-15 (Cont'd.)

AUTOMOTIVE GAS TURBINE SELECTION STUDY
PARTS LIST
PD-1 ENGINE (SINGLE SHAFT)

PART	NAME	DESCRIPTION	MATERIAL	TYPE	COST/LB	WEIGHT	TOTAL WEIGHT	NUMBER OF PARTS	COST PER PART	TOTAL COST
099	DRIVE GEAR	MAIN OIL PUMP	8620 STEEL	FOR	0.30	0.453	0.45	1	0.14	0.14
101	WORM GEAR	-	8620 STEEL	FOR	0.30	0.316	0.63	2	0.09	0.19
102	WORM	-	8620 STEEL	FOR	0.30	0.481	0.96	2	0.14	0.29
109	COUPLING	SPLINED	4340 STEEL	TUB	0.20	0.360	0.72	2	0.07	0.14
111	SPROCKET	REGENERATOR DRIVE	8620 STEEL	FOR	0.30	1.020	2.04	2	0.31	0.61
114	WOODRUFF KEY	REG. DRIVE SHAFT	4340 STEEL	PUR	0.00	0.000	0.00	2	0.03	0.06

TOTAL WEIGHT = 44.92 LBS. TOTAL COST = $ 13.54 NUMBER OF PARTS = 73

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Table A-16

Nodular Iron

AUTOMOTIVE GAS TURBINE SELECTION STUDY
PARTS LIST
PD-1 ENGINE (SINGLE SHAFT)

PART	NAME	DESCRIPTION	MATERIAL	TYPE	COST/LB	WEIGHT	TOTAL WEIGHT	NUMBER OF PARTS	COST PER PART	TOTAL COST
005	COVER PLATE	REGENERATOR	NOD. IRON	CST	0.25	17.000	34.00	2	4.25	8.50
006	COVER PLATE	TOP	NOD. IRON	CST	0.30	10.000	10.00	1	3.00	3.00
013	SUPPORT	MAIN BEARING	NOD. IRON	CST	0.30	1.300	1.30	1	0.39	0.39
023	HOUSING	MAIN BEARING	NOD. IRON	CST	0.25	26.000	26.00	1	6.50	6.50
031	COVER PLATE	FRONT	NOD. IRON	CST	0.25	12.000	12.00	1	3.00	3.00
041	SUP FRAME	CENTER	NOD. IRON	CST	0.20	140.000	140.00	1	28.00	28.00
058	SEAL HOUSING	COMPRESSOR BEARING	NOD. IRON	CST	0.30	0.372	0.37	1	0.11	0.11
063	SHEAVE	V BELT	NOD. IRON	PUR	0.00	0.400	1.20	3	0.26	0.78
083	OIL PUMP	SCAVENGE	NOD. IRON	PUR	0.00	1.580	1.58	1	0.55	0.55
084	COVER	SCAVENGE OIL PUMP	NOD. IRON	PUR	0.00	0.485	0.48	1	0.17	0.17
095	OIL SUMP	-	NOD. IRON	CST	0.30	5.500	5.50	1	1.65	1.65
110	END CAP	-	NOD. IRON	CST	0.30	0.230	0.46	2	0.07	0.14

TOTAL WEIGHT = 232.90 LBS. TOTAL COST = $ 52.79 NUMBER OF PARTS = 16

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Table A-17

304 Stainless Steel

AUTOMOTIVE GAS TURBINE SELECTION STUDY
PARTS LIST
PD-1 ENGINE (SINGLE SHAFT)

PART	NAME	DESCRIPTION	MATERIAL	TYPE	COST/LB	WEIGHT	TOTAL WEIGHT	NUMBER OF PARTS	COST PER PART	TOTAL COST
007	FLANGE	TURBINE INLET REGENERATOR	304 SST	CST	2.00	3.000	3.00	1	6.00	6.00
009	FLANGE	AFT INNER COMBUST SUPPORT	304 SST	CST	2.00	2.170	2.17	1	4.34	4.34
010A	VALVE	COMBUSTOR AIR	304 SST	TUB	0.70	30.432	0.43	1	0.30	0.30
010B	VALVE	COMBUSTOR AIR	304 SST	SHT	0.62	0.432	0.43	1	0.27	0.27
014	SHAFT	REGENERATOR	304 SST	BAR	0.62	0.626	1.25	2	0.39	0.78
015	SPACER	REGENERATOR SHAFT	304 SST	EXT	0.70	0.529	1.06	2	0.37	0.74
016	HOUSING	REGENERATOR SHAFT	304 SST	CST	2.00	1.030	2.06	2	2.06	4.12
020	HEX NUT	REGENERATOR SHAFT	304 SST	BAR	0.62	0.035	0.07	2	0.02	0.04
022	INSUL. SHELL	TURB EXIT&REGEN(IN 25)	304 SST	SHT	0.00	0.000	0.00	1	0.00	0.00
025	SHELL	TURBINE EXHAUST INNER	304 SST	SHT	0.62	2.300	2.30	1	1.43	1.43
026A	SHELL	TURB INLET EXHAUST OUTER	304 SST	PIC	2.00	8.250	8.25	1	16.50	16.50
026C	SHELL	TURB INLET EXHAUST OUTER	304 SST	SHT	0.62	3.250	3.25	1	2.02	2.02
027	LINK	NOZZLE VANE ACTUATOR	304 SST	PIC	2.00	0.084	6.05	72	0.17	12.10
052	TIP SEAL	TURBINE LABYRINTH	304 SST	HNC	21.00	0.050	0.05	1	1.05	1.05
054	ACT RING	TURBINE NOZZLE	304 SST	CST	2.00	3.100	3.10	1	6.20	6.20
055A	LINKAGE ASSY	TURB NOZZLE ACT RING	304 SST	BAR	0.62	0.320	0.64	2	0.20	0.40
055B	LINKAGE ASSY	TURB NOZZLE ACT RING	304 SST	BAR	0.62	0.520	1.04	2	0.32	0.64
055C	LINKAGE ASSY	TURB NOZZLE ACT RING	304 SST	CST	2.00	0.220	0.44	2	0.44	0.88
055D	LINKAGE ASSY	TURB NOZZLE ACT RING	304 SST	PUR	0.00	0.000	0.00	6	0.01	0.06
055E	LINKAGE ASSY	TURB NOZZLE ACT RING	304 SST	PUR	0.00	0.000	0.00	2	0.04	0.08
117	SHELL	TURBINE NOZZLE INNER	304 SST	SHT	0.62	3.500	3.50	1	2.17	2.17

TOTAL WEIGHT = 39.09 LBS. TOTAL COST = $ 60.11 NUMBER OF PARTS = 105

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Table A-18

Aluminum

AUTOMOTIVE GAS TURBINE SELECTION STUDY
PARTS LIST
PD-1 ENGINE (SINGLE SHAFT)

PART	NAME	DESCRIPTION	MATERIAL	TYPE	COST/LB	WEIGHT	TOTAL WEIGHT	NUMBER OF PARTS	COST PER PART	TOTAL COST
002	COMPRESSOR	-	C355-T61 AL	PIC	9.50	2.500	2.50	1	23.75	23.75
029	VANES	COMPRESSOR DIFFUSER	142 AL	DCS	0.45	0.193	5.98	31	0.09	2.69
042	SCROLL	COMPRESSOR OUTER	333 AL	DCS	0.55	10.600	10.60	1	5.83	5.83
043	ACTUATOR RING	DIFFUSER VANE	333 AL	DCS	0.45	1.425	1.42	1	0.64	0.64
075	COVER PLATE	MAIN SHAFT AFT BEARING	43 AL	DCS	0.45	0.234	0.23	1	0.11	0.11
078	SCROLL	COMPRESSOR INNER	43 AL	DCS	0.45	15.000	15.00	1	6.75	6.75
079	COVER PLATE	AFT	43 AL	DCS	0.45	8.100	8.10	1	3.65	3.65
085	BRACKET	DIFFUSER VANE ACT MOTOR	43 AL	DCS	0.45	6.100	6.10	1	2.75	2.75
098	CAN	SPEED REDUCER FRONT HALF	43 AL	DCS	0.45	1.252	2.50	2	0.56	1.13
100	CASE	SPEED REDUCER REAR HALF	33 AL	DCS	0.45	1.174	2.35	2	0.53	1.06

TOTAL WEIGHT = 54.79 LBS. TOTAL COST = $ 48.34 NUMBER OF PARTS = 42

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Table A-19

Precision Investment Castings

AUTOMOTIVE GAS TURBINE SELECTION STUDY
PARTS LIST
PD-1 ENGINE (SINGLE SHAFT)

PART	NAME	DESCRIPTION	MATERIAL	TYPE	COST/LB	WEIGHT	TOTAL WEIGHT	NUMBER OF PARTS	COST PER PART	TOTAL COST
001	TURBINE	-	INC 713LC	PIC	5.50	7.500	7.50	1	41.25	41.25
002	COMPRESSOR	-	C355-T61 AL	PIC	9.50	2.500	2.50	1	23.75	23.75
026A	SHELL	TURB INLET EXHAUST OUTER	304 SST	PIC	2.00	8.250	8.25	1	16.50	16.50
026B	SHELL	TURB INLET EXHAUST OUTER	HAST. X	PIC	6.00	4.500	4.50	1	27.00	27.00
027	LINK	NOZZLE VANE ACTUATOR	304 SST	PIC	2.00	0.084	6.05	72	0.17	12.10
030	VANES	TURBINE STATOR	INC 713LC	PIC	10.00	0.051	5.10	100	0.51	51.00

TOTAL WEIGHT = 33.90 LBS. TOTAL COST = $171.60 NUMBER OF PARTS = 176

Balloon Drawings and Parts Lists, PD-2 Engine - Figures A-4 thru A-7 (Dwg. 221R909) correspond, but are not exactly the same as, Dwg. 221R911. The ballooned numbers identify the parts in the detailed lists.

Table A-20 lists the detailed parts lists in the same fashion as was done for the PD-1 engine.

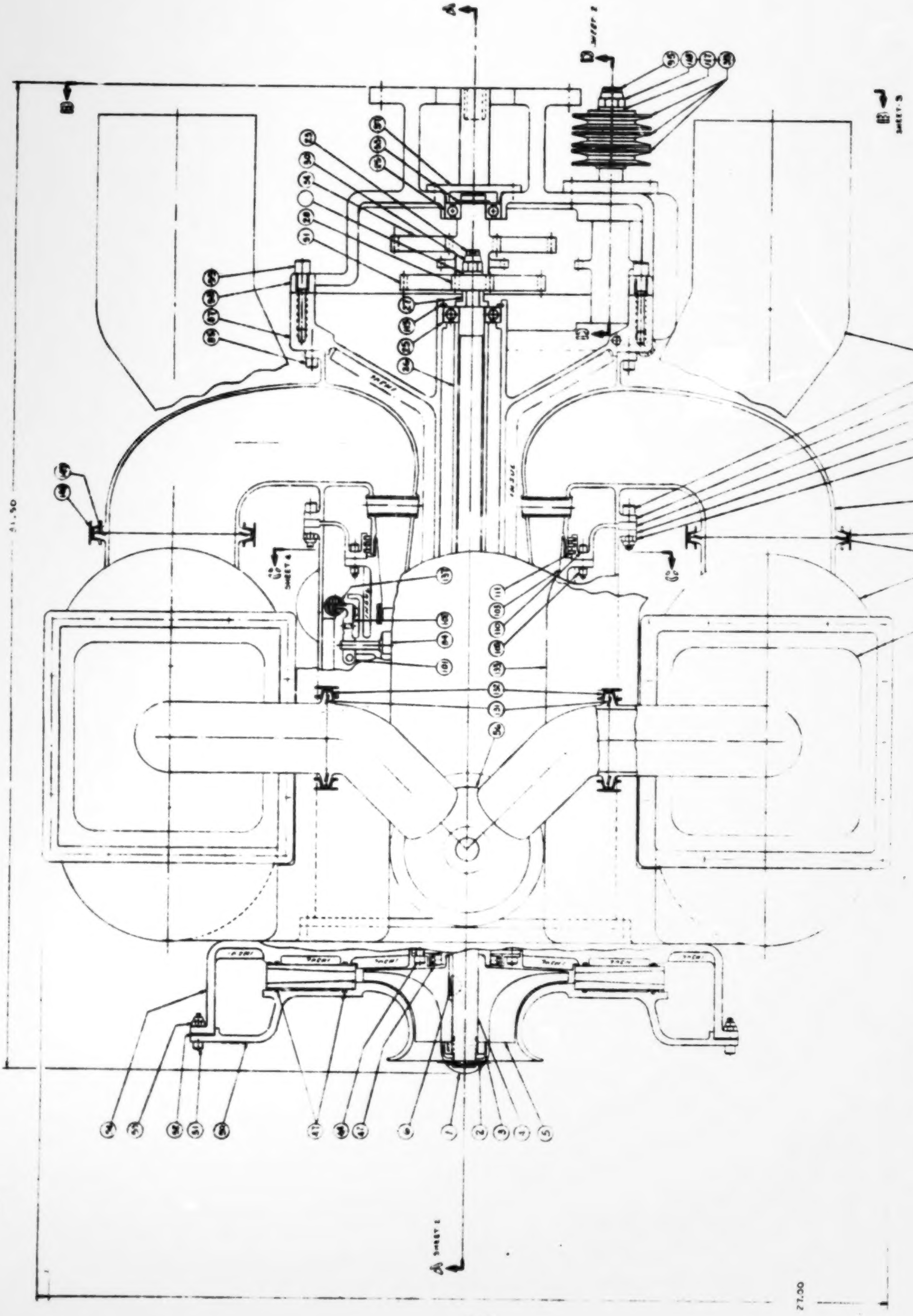
PLAN VIEW
41.50
27.00
A
B
C
D
1
2
3
4
5

Figure A-4. PD-2 Engine Drawing With Part Numbers. Plan View.

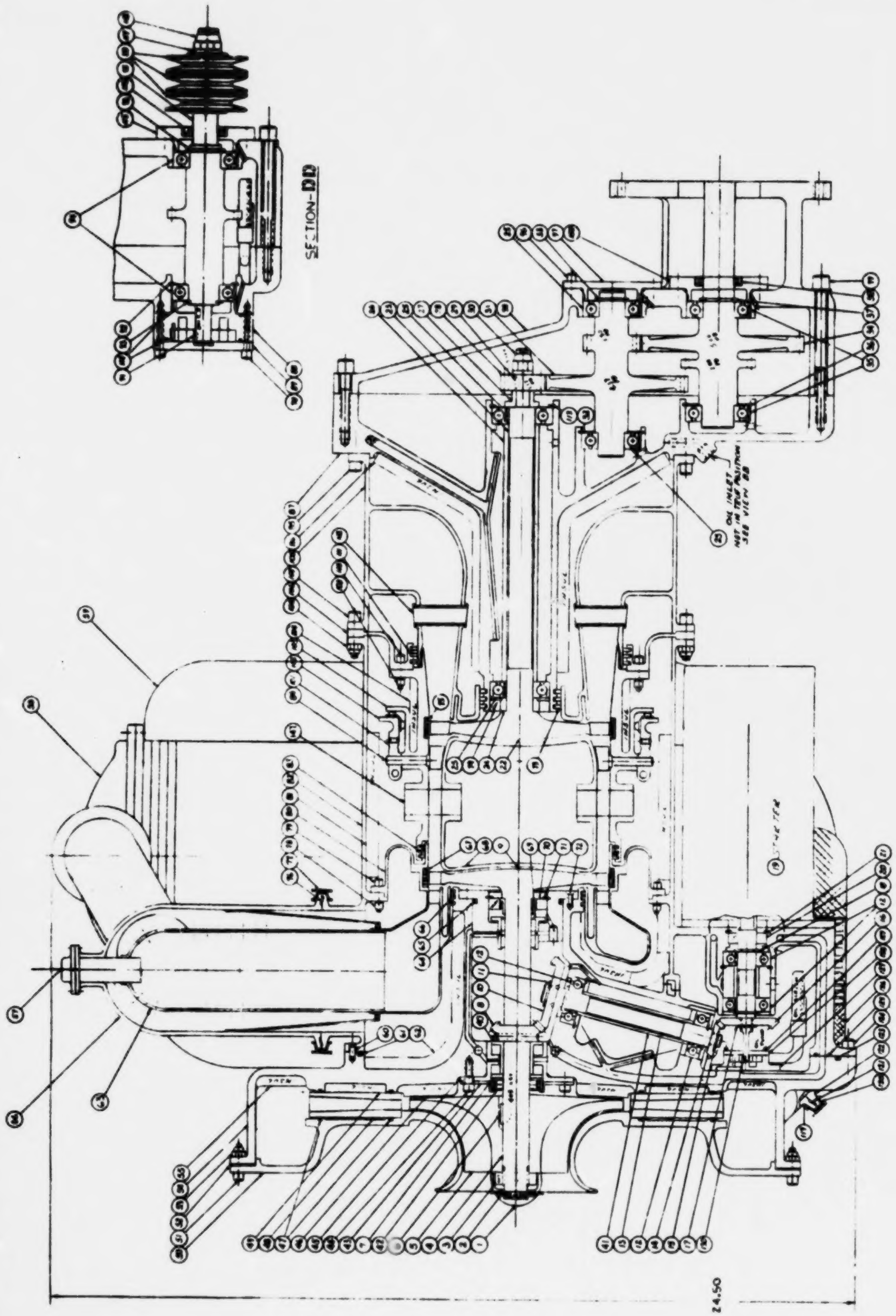
SECTION-DD
24.50

Figure A-5. PD-2 Engine Drawing With Part Numbers. Sections AA and DD.

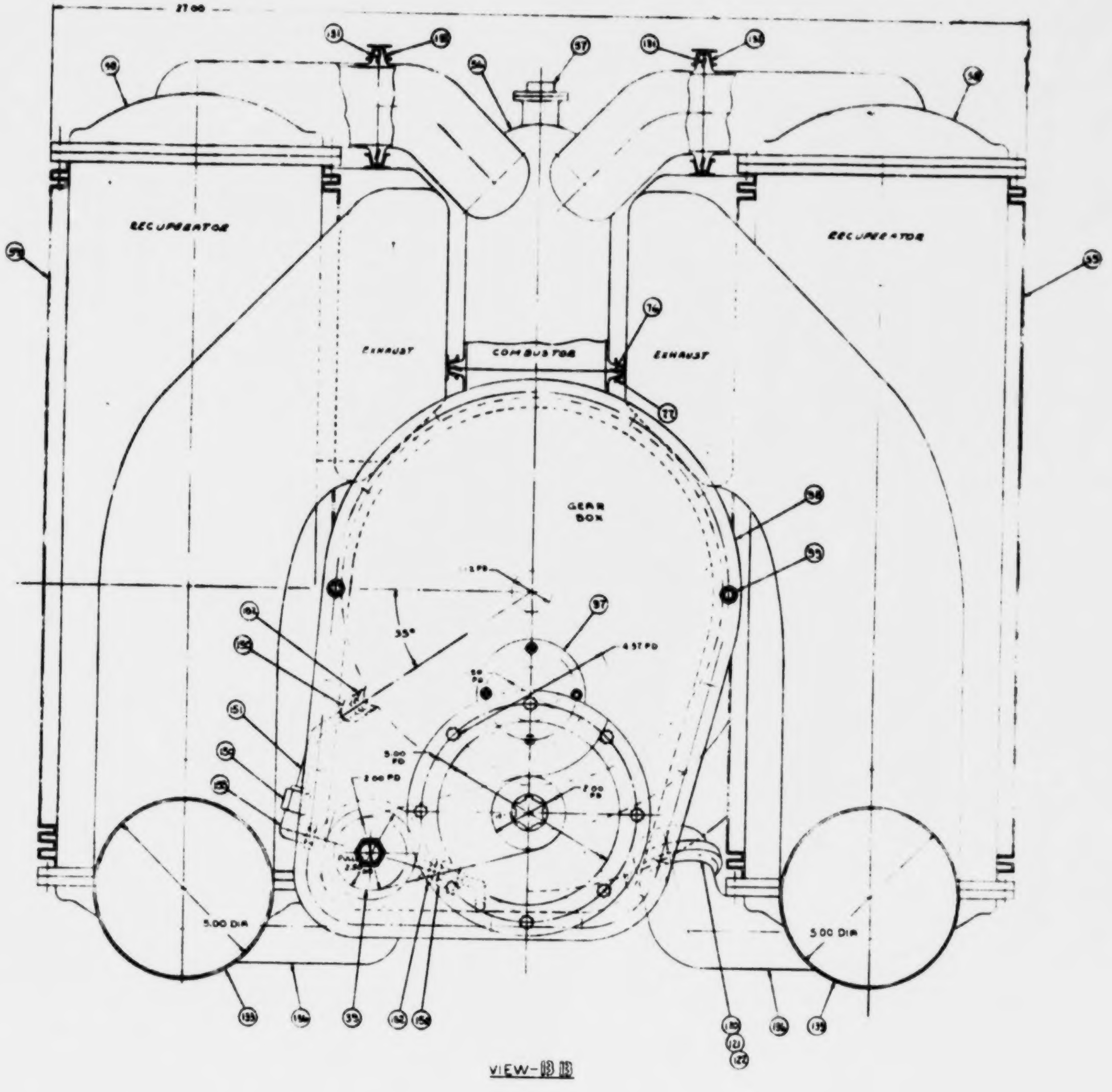
27.00
RECUPERATOR
RECUPERATOR
57
74
EXHAUST
COMBUSTOR
EXHAUST
GEAR BOX
35°
4.57 PD
5.00 PD
2.00 PD
2.00 PD
5.00 DIA
5.00 DIA
37
VIEW-BB

Figure A-6. PD-2 Engine Drawing With Part Numbers. View BB.

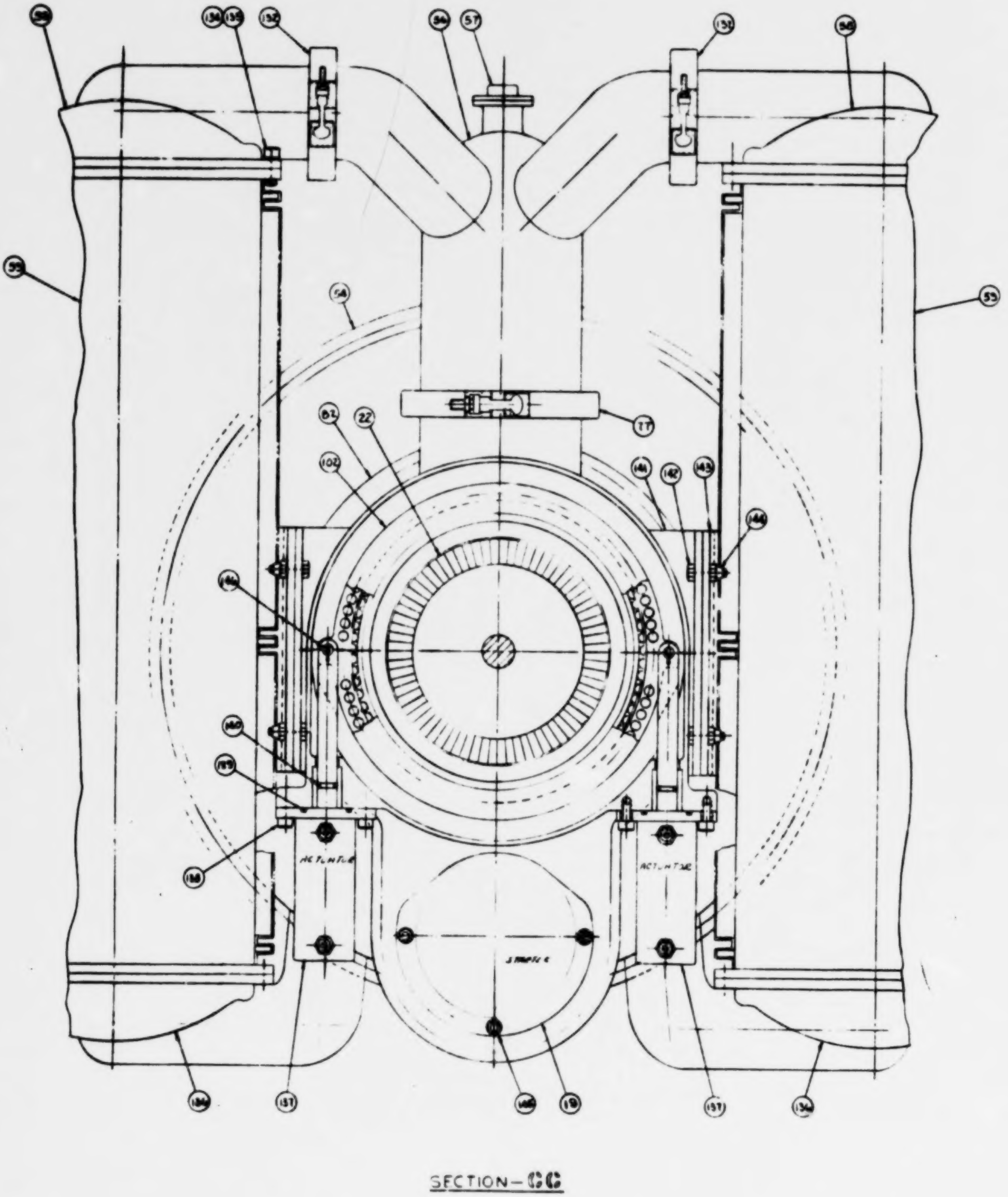
56
134
135
132
54
57
132
56
55
55
58
77
82
22
102
141
142
143
144
144
160
159
158
ACTUATOR
ACTUATOR
136
137
166
19
137
136
SECTION-CC

Figure A-7. PD-2 Engine Drawing With Part Numbers. Section CC

Table A-20

Free Turbine Engine, Parts Breakdown, PD-2

Item	Table No.	Weight (Lbs.)	Cost ($)
Complete Parts List	A-21	425.18	432.32
Castings	A-22	259.78	251.46
Forgings	A-23	9.34	2.80
Bar, Tube & Extrusions	A-24	5.82	5.56
Sheet & Plate	A-25	66.50	39.47
Purchased Parts	A-26	83.33	122.64
Miscellaneous	A-27	0.40	10.39
Hastelloy X and Inconel 713LC	A-28	10.71	98.06
Steel	A-29	30.34	7.27
Nodular Iron	A-30	175.63	47.06
304 Stainless Steel	A-31	102.56	137.69
Aluminum	A-32	0.29	0.30
Precision Investment Castings	A-33	4.29	20.20

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Table A-21

Complete Parts List

AUTOMOTIVE GAS TURBINE SELECTION STUDY
PARTS LIST
PD-2 ENGINE (FREE TURBINE)

PART	NAME	DESCRIPTION	MATERIAL	TYPE	COST/LB	WEIGHT	TOTAL WEIGHT	NUMBER OF PARTS	COST PER PART	TOTAL COST
001	BULLETNOSE	COMPRESSOR	355 AL	CST	0.45	0.149	0.15	1	0.07	0.07
002	PIN	COMP BULLETNOSE	4140 STEEL	PUR	0.00	0.004	0.00	1	0.01	0.01
003	NUT	COMPRESSOR	4140 STEEL	BAR	0.20	0.089	0.09	1	0.02	0.02
004	SHAFT	GAS GENERATOR	4340 STEEL	BAR	0.20	1.372	1.37	1	0.27	0.27
005	COMPRESSOR	-	410 SST	CST	2.00	3.467	3.47	1	6.93	6.93
006	WOODRUFF KEY	COMPRESSOR	4140 STEEL	PUR	0.00	0.011	0.01	1	0.03	0.03
007	SEAL RING	COMPRESSOR	4140 STEEL	BAR	0.20	0.050	0.05	1	0.01	0.01
008	GEAR	GAS GEN SHAFT	8620 STEEL	FOR	0.30	0.153	0.15	1	0.05	0.05
009	TURBINE	GAS GEN	INC 713LC	PIC	6.50	2.230	2.23	1	14.50	14.50
010	GEAR	STARTER GAS GEN	8620 STEEL	FOR	0.30	0.300	0.30	1	0.09	0.09
011	SNAP RING	STARTER GAS GEN GEAR SHAFT	-	PUR	0.00	0.006	0.02	3	0.04	0.12
012	BEARING	STARTER GAS GEN GEAR SHAFT	-	PUR	0.00	0.122	0.49	4	2.80	11.20
013	GEAR SHAFT	STARTER GAS GEN	4340 STEEL	BAR	0.20	0.340	0.34	1	0.07	0.07
014	GEAR	STARTER GAS GEN	8620 STEEL	FOR	0.30	0.052	0.05	1	0.02	0.02
015	SNAP RING	STARTER GAS GEN GEAR	-	PUR	0.00	0.003	0.00	1	0.04	0.04
016	GEAR	STARTER	8620 STEEL	FOR	0.30	0.205	0.21	1	0.06	0.06
017	SNAP RING	STARTER GEAR	-	PUR	0.00	0.003	0.00	1	0.04	0.04
018	SNAP RING	STARTER SHAFT BEARING	-	PUR	0.00	0.005	0.01	2	0.04	0.08
020	SEAL	STARTER SHAFT	-	PUR	0.00	0.058	0.06	1	0.10	0.10
021	SNAP RING	STARTER SHAFT SEAL	-	PUR	0.00	0.012	0.01	1	0.04	0.04
022	TURBINE	POWER	CMR-60	PIC	2.77	2.061	2.06	1	5.71	5.71
023	SHAFT	POWER TURBINE	4340 STEEL	BAR	0.20	1.257	1.26	1	0.25	0.25
024	SEAL RING	POWER TURBINE SHAFT	4140 STEEL	BAR	0.20	0.070	0.07	1	0.01	0.01
025	BEARING	POWER TURBINE SHAFT	-	PUR	0.00	0.280	1.12	4	2.80	11.20
026	SPACER	POWER TURB SHAFT BEARING	4140 STEEL	TUB	0.20	0.893	0.89	1	0.18	0.18
027	RETAINER	AFT POWER TURB SHAFT BEAR	4140 STEEL	BAR	0.20	0.052	0.05	1	0.01	0.01
028	GEAR	POWER TURB DRIVE	8620 STEEL	FOR	0.30	0.122	0.12	1	0.04	0.04
029	WOODRUFF KEY	POWER TURB DRIVE GEAR	-	PUR	0.00	0.001	0.00	1	0.03	0.03
030	NUT	POWER TURBINE SHAFT	4140 STEEL	BAR	0.20	0.027	0.03	1	0.01	0.01
031	GEAR ASSY	REDUCTION	8620 STEEL	FOR	0.30	2.680	2.68	1	0.80	0.80
032	SNAP RING	FRONT RED GEAR BEAR	-	PUR	0.00	0.011	0.01	1	0.04	0.04
033	SNAP RING	REAR RED GEAR BEAR	-	PUR	0.00	0.008	0.01	1	0.04	0.04
034	GEAR ASSY	ACCESSORY DRIVE	8620 STEEL	FOR	0.30	4.230	4.23	1	1.27	1.27

Table A-21 (Cont'd.)

AUTOMOTIVE GAS TURBINE SELECTION STUDY
PARTS LIST
PD-2 ENGINE (FREE TURBINE)

PART	NAME	DESCRIPTION	MATERIAL	TYPE	COST/LB	WEIGHT	TOTAL WEIGHT	NUMBER OF PARTS	COST PER PART	TOTAL COST
035	BEARING	ACCESSORY DRIVE GEAR	-	PUR	0.00	0.339	0.68	2	2.80	5.60
036	SNAP RING	ACCESSORY DRIVE GEAR	-	PUR	0.00	0.012	0.01	1	0.04	0.04
037	SNAP RING	-	-	PUR	0.00	0.012	0.04	3	0.04	0.12
038	SEAL	ACCESSORY DRIVE GEAR SHAFT	-	PUR	0.00	0.026	0.03	1	0.10	0.10
039	SHEAVE	V BELT	NOD. IRON	PUR	0.00	0.477	1.43	3	0.26	0.78
040	BEARING	THRUST PWR GEN SHAFT	-	PUR	0.00	0.113	0.11	1	2.00	2.00
041	SPACER	GAS GEN GEAR SHAFT BRGS	4140 STEEL	TUB	0.20	0.170	0.17	1	0.03	0.03
042	OIL SEAL	FRONT GAS GEN SHAFT	-	PUR	0.00	0.106	0.11	1	0.10	0.10
043	RETAINER	GAS GEN SHAFT FNT OIL SEAL	4140 STEEL	BAR	0.20	0.141	0.14	1	0.03	0.03
044	SCREW	-	-	PUR	0.00	0.006	0.05	8	0.01	0.08
045	HOUSING	BEARING	NOD. IRON	PUR	0.00	0.745	0.74	1	0.50	0.50
046	INSULATION	-	-	PUR	0.20	1.045	1.05	1	0.21	0.21
047	O RING	COMP DIFFUSER	-	PUR	0.00	0.068	0.14	2	0.03	0.06
048	DIFFUSER	COMPRESSOR	NOD. IRON	CST	0.30	8.726	8.73	1	2.62	2.62
049	INSULATION	-	-	PUR	0.20	1.576	1.58	1	0.32	0.32
050	COVER PLATE	FRONT	NOD. IRON	CST	0.25	18.985	18.98	1	4.75	4.75
051	SCREW	-	-	PUR	0.00	0.006	0.07	12	0.01	0.12
052	GASKET	-	PAPER	PUR	0.00	0.081	0.08	1	0.10	0.10
053	NUT	-	-	PUR	0.00	0.005	0.07	12	0.01	0.12
054	HOUSING	BEARING GAS GEN	NOD. IRON	CST	0.25	41.384	41.38	1	10.35	10.35
055	INSULATION	-	-	PUR	0.20	2.090	2.09	1	0.42	0.42
056	OUTER SHELL	COMBUSTOR	304 SST	CST	2.00	7.213	7.21	1	14.43	14.43
057	CAP	FUEL INJECTOR	HAST. X	CST	10.00	0.195	0.20	1	1.95	1.95
058	COVER	RECUPERATOR TOP	NOD. IRON	CST	0.30	7.533	15.07	2	2.26	4.52
059A	RECUPERATOR	(SHELL)	304 SST	SHT	0.70	26.172	52.34	2	18.32	36.64
059B	RECUPERATOR	(EXHAUST PIPE)	AL. STEEL	SHT	0.20	7.080	14.16	2	1.42	2.83
059C	RECUPERATOR	(CORE)	CERVIT	PUR	0.00	21.728	43.46	2	30.00	60.00
060	DOWEL PIN	-	-	PUR	0.00	0.002	0.00	1	0.01	0.01
061	O RING	-	-	PUR	0.00	0.104	0.10	1	0.03	0.03
062	INSULATION	-	-	PUR	0.20	4.440	4.44	1	0.89	0.89
063	LINER	COMBUSTOR	HAST. X	EXT	3.50	0.751	0.75	1	2.63	2.63
064	HOUSING	BEARING	NOD. IRON	PUR	0.00	1.549	1.55	1	0.50	0.50
065	SEAL	METALLIC	HS-25	PUR	0.00	0.053	0.05	1	0.25	0.25

Table A-21 (Cont'd.)

AUTOMOTIVE GAS TURBINE SELECTION STUDY
PARTS LIST
PD-2 ENGINE (FREE TURBINE)

PART	NAME	DESCRIPTION	MATERIAL	TYPE	COST/LB	WEIGHT	TOTAL WEIGHT	NUMBER OF PARTS	COST PER PART	TOTAL COST
066	RING SEAL	GAS GEN TURBINE	HS-25	PUR	0.00	0.028	0.08	3	0.15	0.45
067	TIP SEAL	GAS GEN TURBINE	HAST. X	HNC	70.80	0.122	0.12	1	8.64	8.64
068	CASING	TURBINE INNER	304 SST	CST	2.00	2.891	2.89	1	5.78	5.78
069	BEARING	REAR GAS GEN	-	PUR	0.00	0.196	0.20	1	2.00	2.00
070	OIL SEAL	GAS GEN TURBINE	-	PUR	0.00	0.204	0.20	1	0.10	0.10
071	SNAP RING	GAS GEN TURB OIL SEAL	-	PUR	0.00	0.017	0.02	1	0.04	0.04
072	SCREW	-	-	PUR	0.00	0.003	0.02	8	0.01	0.08
073	RING SEAL	POWER TURBINE	302 SST	PUR	0.00	0.023	0.07	3	0.10	0.30
074	OIL SEAL	POWER TURBINE	-	PUR	0.00	0.076	0.08	1	0.10	0.10
075	TIP SEAL	POWER TURBINE	304 SST	HNC	10.00	0.130	0.13	1	1.30	1.30
076	GASKET	SEAL	PAPER	PUR	0.00	0.039	0.04	1	0.10	0.10
077	BOLT	-	304 SST	PUR	0.00	0.034	0.27	8	0.01	0.08
078	NUT	-	304 SST	PUR	0.00	0.004	0.05	12	0.01	0.12
079	SCREW	-	304 SST	PUR	0.00	0.004	0.05	12	0.01	0.12
080A	NOZ. HOUSING	GAS GEN TURBINE	HAST. X	CST	10.00	6.832	6.83	1	68.32	68.32
080B	NOZ. HOUSING	GAS GEN TURBINE	HAST. X	EXT	3.50	0.580	0.58	1	2.03	2.03
081	INSULATION	-	-	PUR	0.20	4.739	4.74	1	0.95	0.95
082	CASING	OUTER	B50T13	CST	0.30	22.350	22.35	1	6.70	6.70
083	RING SEAL	GAS GEN NOZZLE HOUSING	HS-25	-	0.00	0.048	0.14	3	0.15	0.45
084	NOZZLE VANE	POWER TURBINE	CRM-60	CST	2.77	0.029	1.04	36	0.08	2.89
085	BOLTS	-	4140 STEEL	PUR	0.00	0.007	0.21	30	0.01	0.30
086	INSULATION	POWER TURB BEAR HOUSING	-	PUR	0.20	3.890	3.89	1	0.78	0.78
087	HOUSING	BEARING POWER TURBINE	NOD. IRON	CST	0.25	28.776	28.78	1	7.19	7.19
088	OIL PUMP	-	-	PUR	0.00	1.269	1.27	1	2.15	2.15
089	COVER	OIL PUMP	NOD. IRON	CST	0.30	0.517	0.52	1	0.16	0.16
090	BOLTS	-	-	PUR	0.00	0.002	0.02	8	0.01	0.08
091	WOODRUFF KEY	OIL PUMP DRIVE SHAFT	-	PUR	0.00	0.001	0.00	1	0.03	0.03
092	SNAP RING	OIL PUMP BEAR OUTER	-	PUR	0.00	0.018	0.02	1	0.04	0.04
093	SNAP RING	OIL PUMP BEAR INNER	-	PUR	0.00	0.009	0.02	2	0.04	0.08
094	BEARING	OIL PUMP	-	PUR	0.00	0.339	0.68	2	2.80	5.60
095	DRIVE SHAFT	OIL PUMP	4140 STEEL	FOR	0.30	1.596	1.60	1	0.48	0.48
096	BOLT	-	-	PUR	0.00	0.002	0.05	24	0.01	0.24
097	COVER	REDUCTION GEAR BEARING	NOD. IRON	CST	0.30	0.430	0.43	1	0.13	0.13

Table A-21 (Cont'd.)

AUTOMOTIVE GAS TURBINE SELECTION STUDY
PARTS LIST
PD-2 ENGINE (FREE TURBINE)

PART	NAME	DESCRIPTION	MATERIAL	TYPE	COST/LB	WEIGHT	TOTAL WEIGHT	NUMBER OF PARTS	COST PER PART	TOTAL COST
098	COVER	GEAR BOX	NOD. IRON	CST	0.25	36.632	36.63	1	9.16	9.16
099	BOLT	-	-	PUR	0.00	0.017	0.41	24	0.01	0.24
100	COVER	ACC DRV SHAFT REAR BEAR	NOD. IRON	CST	0.30	1.217	1.22	1	0.37	0.37
101	ACTUATOR LINK	POWER TURB NOZZLE	304 SST	CST	2.00	0.049	1.76	36	0.10	3.53
102	ACTUATOR RING	POWER TURB NOZZLE	304 SST	CST	2.00	2.240	2.24	1	4.48	4.48
103	CASING	POWER TURB OUTER	304 SST	CST	2.00	11.481	11.48	1	22.96	22.96
104	INSULATION	-	-	PUR	0.20	1.125	1.13	1	0.22	0.22
105	NUT	-	-	PUR	0.00	0.012	0.10	8	0.01	0.08
106	HOUSING	EXHAUST SCROLL RING SEAL	NOD. IRON	CST	0.30	3.833	3.83	1	1.15	1.15
107	BOLT	-	304 SST	PUR	0.00	0.012	0.10	8	0.01	0.08
108	SCROLL	EXHAUST	304 SST	CST	2.00	21.702	21.70	1	43.40	43.40
109	NUT	-	304 SST	PUR	0.00	0.003	0.04	12	0.01	0.12
110	SCREW	-	304 SST	PUR	0.00	0.003	0.04	12	0.01	0.12
111	RING SEAL	EXHAUST SCROLL	INC. X	PUR	0.00	0.059	0.18	3	0.20	0.60
112	STRUT	TURBINE EXHAUST	304 SST	CST	2.00	0.191	0.76	4	0.38	1.53
113	SNAP RING	POWER TURB REAR BRG	-	PUR	0.00	0.022	0.02	1	0.04	0.04
114	O RING	OIL PUMP	NEOPRENE	PUR	0.00	0.016	0.03	2	0.03	0.06
115	COVER	OIL PUMP SHAFT REAR BRG	NOD. IRON	CST	0.30	1.217	1.22	1	0.37	0.37
116	OIL SEAL	OIL PUMP SHAFT REAR BRG	-	PUR	0.00	0.018	0.02	1	0.10	0.10
117	LOCK WASHER	OIL PUMP SHAFT	-	PUR	0.00	0.007	0.01	1	0.04	0.04
118	NUT	OIL PUMP SHAFT	-	PUR	0.00	0.022	0.02	1	0.01	0.01
119	FLANGE	MARMAN	-	-	0.00	0.000	0.00	0	0.00	0.00
120	MARMAN CLAMP	-	-	PUR	0.00	0.457	0.46	1	1.00	1.00
121	SEAL RING	MARMAN FLANGE	-	PUR	0.00	0.023	0.02	1	0.10	0.10
122	MARMAN FLANGE	-	-	-	0.00	0.000	0.00	0	0.00	0.00
123	O RING	-	NEOPRENE	PUR	0.00	0.212	0.21	1	0.03	0.03
124	BOLT	-	-	PUR	0.00	0.049	1.18	24	0.01	0.24
125	O RING	-	NEOPRENE	PUR	0.00	0.003	0.00	1	0.03	0.03
126	COVER	BOTTOM RECUP	NOD. IRON	CST	0.30	7.533	15.07	2	2.26	4.52
127	OIL PUMP	SCAVENGE	-	PUR	0.00	0.255	0.25	1	0.55	0.55
128	COVER	SCAVENGE OIL PUMP	NOD. IRON	CST	0.30	0.051	0.05	1	0.02	0.02
129	SCREWS	-	-	PUR	0.00	0.001	0.01	8	0.01	0.08
130	WOODRUFF KEY	SCAVENGE OIL PUMP SHAFT	-	PUR	0.00	0.003	0.00	1	0.03	0.03

Table A-21 (Cont'd.)

AUTOMOTIVE GAS TURBINE SELECTIJN STUDY
PARTS LIST
PD-2 ENGINE (FREE TURBINE)

PART	NAME	DESCRIPTION	MATERIAL	TYPE	COST/LB	WEIGHT	TOTAL WEIGHT	NUMBER OF PARTS	COST PER PART	TOTAL COST
131	GASKET	-	-	PUR	0.00	0.023	0.05	2	0.10	0.20
132	MARMAN CLAMP	-	-	PUR	0.00	0.274	0.55	2	1.00	2.00
134	LOCK WASHER	-	-	PUR	0.00	0.001	0.05	60	0.01	0.30
135	SCREW	-	-	PUR	0.00	0.010	0.60	60	0.01	0.30
137	ACTUATOR	-	-	PUR	0.00	1.524	3.05	2	2.00	4.00
138	SCREW	-	-	PUR	0.00	0.008	0.06	8	0.01	0.08
139	O RING	-	NEOPRENE	PUR	0.00	0.005	0.01	2	0.03	0.06
140	O RING	-	NEOPRENE	PUR	0.00	0.002	0.00	2	0.03	0.06
141	BRACKET	-	1010 STEEL	PUR	0.00	0.509	1.02	2	0.10	0.20
142	BOLT	-	-	PUR	0.00	0.006	0.05	8	0.01	0.08
143	BRACKET	-	1010 STEEL	PUR	0.00	0.569	1.14	2	0.10	0.20
144	NUT	-	-	PUR	0.00	0.008	0.06	8	0.01	0.08
145	SCREW	SOC HEAD	-	PUR	0.00	0.106	0.32	3	0.01	0.03
147	TUBE	STRUT	304 SST	CST	2.00	0.375	1.50	4	0.75	3.00
148	MARMAN CLAMP	-	-	PUR	0.00	0.231	0.46	2	1.00	2.00
149	GASKET	-	-	-	0.00	0.000	0.00	2	0.00	0.00
150	FITTING	TUBE	AL	PUR	0.00	0.009	0.03	3	0.05	0.15
151	TUBE	1/4 IN OD	AL	EXT	0.40	0.028	0.03	1	0.01	0.01
152	BUSHING	ADAPTER	AL	PUR	0.00	0.035	0.07	2	0.01	0.02
153	ELBOW FITTING	-	AL	PUR	0.00	0.019	0.02	1	0.05	0.05
154	LOCK WASHER	-	-	PUR	0.00	0.008	0.01	1	0.01	0.01

TOTAL WEIGHT = 425.18 LBS. TOTAL COST = $432.32 NUMBER OF PARTS = 624

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Table A-22

Castings

AUTOMOTIVE GAS TURBINE SELECTION STUDY
PARTS LIST
PD-2 ENGINE (FREE TURBINE)

PART	NAME	DESCRIPTION	MATERIAL	TYPE	COST/LB	WEIGHT	TOTAL WEIGHT	NUMBER OF PARTS	COST PER PART	TOTAL COST
001	BULLETNOSE	COMPRESSOR	355 AL	CST	0.45	0.149	0.15	1	0.07	0.07
005	COMPRESSOR	-	410 SST	CST	2.00	3.467	3.47	1	6.93	6.93
009	TURBINE	GAS GEN	INC 713LC	PIC	6.50	2.230	2.23	1	14.50	14.50
022	TURBINE	POWER	CMR-60	PIC	2.77	2.061	2.06	1	5.71	5.71
048	DIFFUSER	COMPRESSOR	NOD. IRON	CST	0.30	8.726	8.73	1	2.62	2.62
050	COVER PLATE	FRONT	NOD. IRON	CST	0.25	18.985	18.98	1	4.75	4.75
054	HOUSING	BEARING GAS GEN	NOD. IRON	CST	0.25	41.384	41.38	1	10.35	10.35
056	OUTER SHELL	COMBUSTOR	304 SST	CST	2.00	7.213	7.21	1	14.43	14.43
057	CAP	FUEL INJECTOR	HAST. X	CST	10.00	0.195	0.20	1	1.95	1.95
058	COVER	RECUPERATOR TOP	NOD. IRON	CST	0.30	7.533	15.07	2	2.26	4.52
068	CASING	TURBINE INNER	304 SST	CST	2.00	2.891	2.89	1	5.78	5.78
080A	NOZ. HOUSING	GAS GEN TURBINE	HAST. X	CST	10.00	6.832	6.83	1	68.32	68.32
082	CASING	OUTER	B50T13	CST	0.30	22.350	22.35	1	6.70	6.70
084	NOZZLE VANE	POWER TURBINE	CRM-60	CST	2.77	0.029	1.04	36	0.08	2.89
087	HOUSING	BEARING POWER TURBINE	NOD. IRON	CST	0.25	28.776	28.78	1	7.19	7.19
089	COVER	OIL PUMP	NOD. IRON	CST	0.30	0.517	0.52	1	0.16	0.16
097	COVER	REDUCTION GEAR BEARING	NOD. IRON	CST	0.30	0.430	0.43	1	0.13	0.13
098	COVER	GEAR BOX	NOD. IRON	CST	0.25	36.632	36.63	1	9.16	9.16
100	COVER	ACC DRV SHAFT REAR BEAR	NOD. IRON	CST	0.30	1.217	1.22	1	0.37	0.37
101	ACTUATOR LINK	POWER TURB NOZZLE	304 SST	CST	2.00	0.049	1.76	36	0.10	3.53
102	ACTUATOR RING	POWER TURB NOZZLE	304 SST	CST	2.00	2.240	2.24	1	4.48	4.48
103	CASING	POWER TURB OUTER	304 SST	CST	2.00	11.481	11.48	1	22.96	22.96
106	HOUSING	EXHAUST SCROLL RING SEAL	NOD. IRON	CST	0.30	3.833	3.83	1	1.15	1.15
108	SCROLL	EXHAUST	304 SST	CST	2.00	21.702	21.70	1	43.40	43.40
112	STRUT	TURBINE EXHAUST	304 SST	CST	2.00	0.191	0.76	4	0.38	1.53
115	COVER	OIL PUMP SHAFT REAR BRG	NOD. IRON	CST	0.30	1.217	1.22	1	0.37	0.37
126	COVER	BOTTOM RECUP	NOD. IRON	CST	0.30	7.533	15.07	2	2.26	4.52
128	COVER	SCAVENGE OIL PUMP	NOD. IRON	CST	0.30	0.051	0.05	1	0.02	0.02
147	TUBE	STRUT	304 SST	CST	2.00	0.375	1.50	4	0.75	3.00

TOTAL WEIGHT = 259.78 LBS.   TOTAL COST = $251.46   NUMBER OF PARTS = 107

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Table A-23

Forgings

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                              AUTOMOTIVE GAS TURBINE SELECTION STUDY
                                        PARTS LIST
                                    PD-2 ENGINE (FREE TURBINE)
PART  NAME         DESCRIPTION              MATERIAL     TYPE COST/LB WEIGHT  TOTAL   NUMBER     COST      TOTAL
                                                                              WEIGHT  OF PARTS  PER PART   COST

008  GEAR          GAS GEN SHAFT            8620 STEEL   FOR   0.30   0.153   0.15      1        0.05      0.05
010  GEAR          STARTER GAS GEN          8620 STEEL   FOR   0.30   0.300   0.30      1        0.09      0.09
014  GEAR          STARTER GAS GEN          8620 STEEL   FOR   0.30   0.052   0.05      1        0.02      0.02
016  GEAR          STARTER                  8620 STEEL   FOR   0.30   0.205   0.21      1        0.06      0.06
028  GEAR          POWER TURB DRIVE         8620 STEEL   FOR   0.30   0.122   0.12      1        0.04      0.04
031  GEAR ASSY     REDUCTION                8620 STEEL   FOR   0.30   2.680   2.68      1        0.80      0.80
034  GEAR ASSY     ACCESSORY DRIVE          8620 STEEL   FOR   0.30   4.230   4.23      1        1.27      1.27
095  DRIVE SHAFT   OIL PUMP                 4140 STEEL   FOR   0.30   1.596   1.60      1        0.48      0.48

TOTAL WEIGHT =    9.34 LBS.   TOTAL COST = $  2.80      NUMBER OF PARTS =   8
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Table A-24

Bar, Tube & Extrusions

AUTOMOTIVE GAS TURBINE SELECTION STUDY
PARTS LIST
PD-2 ENGINE (FREE TURBINE)

PART	NAME	DESCRIPTION	MATERIAL	TYPE	COST/LB	WEIGHT	TOTAL WEIGHT	NUMBER OF PARTS	COST PER PART	TOTAL COST
003	NUT	COMPRESSOR	4140 STEEL	BAR	0.20	0.089	0.09	1	0.02	0.02
004	SHAFT	GAS GENERATOR	4340 STEEL	BAR	0.20	1.372	1.37	1	0.27	0.27
007	SEAL RING	COMPRESSOR	4140 STEEL	BAR	0.20	0.050	0.05	1	0.01	0.01
013	GEAR SHAFT	STARTER GAS GEN	4340 STEEL	BAR	0.20	0.340	0.34	1	0.07	0.07
023	SHAFT	POWER TURBINE	4340 STEEL	BAR	0.20	1.257	1.26	1	0.25	0.25
024	SEAL RING	POWER TURBINE SHAFT	4140 STEEL	BAR	0.20	0.070	0.07	1	0.01	0.01
026	SPACER	POWER TURB SHAFT BEARING	4140 STEEL	TUB	0.20	0.893	0.89	1	0.18	0.18
027	RETAINER	AFT POWER TURB SHAFT BEAR	4140 STEEL	BAR	0.20	0.052	0.05	1	0.01	0.01
030	NUT	POWER TURBINE SHAFT	4140 STEEL	BAR	0.20	0.027	0.03	1	0.01	0.01
041	SPACER	GAS GEN GEAR SHAFT BRGS	4140 STEEL	TUB	0.20	0.170	0.17	1	0.03	0.03
043	RETAINER	GAS GEN SHAFT FNT OIL SEAL	4140 STEEL	BAR	0.20	0.141	0.14	1	0.03	0.03
063	LINER	COMBUSTOR	HAST. X	EXT	3.50	0.751	0.75	1	2.63	2.63
080B	NUZZLE HOUSIN	GAS GEN TURBINE	HAST. X	EXT	3.50	0.580	0.58	1	2.03	2.03
151	TUBE	1/4 IN OD	AL	EXT	0.40	0.028	0.03	1	0.01	0.01

TOTAL WEIGHT = 5.82 LBS. TOTAL COST = $ 5.56 NUMBER OF PARTS = 14

Table A-25

Sheet & Plate

AUTOMOTIVE GAS TURBINE SELECTION STUDY
PARTS LIST
PD-2 ENGINE (FREE TURBINE)

PART	NAME	DESCRIPTION	MATERIAL	TYPE	COST/LB	WEIGHT	TOTAL WEIGHT	NUMBER OF PARTS	COST PER PART	TOTAL COST
059A	RECUPERATOR	(SHELL)	304 SST	SHT	0.70	26.172	52.34	2	18.32	36.64
059B	RECUPERATOR	(EXHAUST PIPE)	AL. STEEL	SHT	0.20	7.080	14.16	2	1.42	2.83

TOTAL WEIGHT = 66.50 LBS. TOTAL COST = $ 39.47 NUMBER OF PARTS = 4

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Table A-26

Purchased Parts

AUTOMOTIVE GAS TURBINE SELECTION STUDY
PARTS LIST
PD-2 ENGINE (FREE TURBINE)

PART	NAME	DESCRIPTION	MATERIAL	TYPE	COST/LB	WEIGHT	TOTAL WEIGHT	NUMBER OF PARTS	COST PER PART	TOTAL COST
002	PIN	COMP BULLETNOSE	4140 STEEL	PUR	0.00	0.004	0.00	1	0.01	0.01
006	WOODRUFF KEY	COMPRESSOR	4140 STEEL	PUR	0.00	0.011	0.01	1	0.03	0.03
011	SNAP RING	STARTER GAS GEN GEAR SHAFT	-	PUR	0.00	0.006	0.02	3	0.04	0.12
012	BEARING	STARTER GAS GEN GEAR SHAFT	-	PUR	0.00	0.122	0.49	4	2.80	11.20
015	SNAP RING	STARTER GAS GEN GEAR	-	PUR	0.00	0.003	0.00	1	0.04	0.04
017	SNAP RING	STARTER GEAR	-	PUR	0.00	0.003	0.00	1	0.04	0.04
018	SNAP RING	STARTER SHAFT BEARING	-	PUR	0.00	0.005	0.01	2	0.04	0.08
020	SEAL	STARTER SHAFT	-	PUR	0.00	0.058	0.06	1	0.10	0.10
021	SNAP RING	STARTER SHAFT SEAL	-	PUR	0.00	0.012	0.01	1	0.04	0.04
025	BEARING	POWER TURBINE SHAFT	-	PUR	0.00	0.280	1.12	4	2.80	11.20
029	WOODRUFF KEY	POWER TURB DRIVE GEAR	-	PUR	0.00	0.001	0.00	1	0.03	0.03
032	SNAP RING	FRONT RED GEAR BEAR	-	PUR	0.00	0.011	0.01	1	0.04	0.04
033	SNAP RING	REAR RED GEAR BEAR	-	PUR	0.00	0.008	0.01	1	0.04	0.04
035	BEARING	ACCESSORY DRIVE GEAR	-	PUR	0.00	0.339	0.68	2	2.80	5.60
036	SNAP RING	ACCESSORY DRIVE GEAR	-	PUR	0.00	0.012	0.01	1	0.04	0.04
037	SNAP RING	-	-	PUR	0.00	0.012	0.04	3	0.04	0.12
038	SEAL	ACCESSORY DRIVE GEAR SHAFT	-	PUR	0.00	0.026	0.03	1	0.10	0.10
039	SHEAVE	V BELT	NOD. IRON	PUR	0.00	0.477	1.43	3	0.26	0.78
040	BEARING	THRUST PWR GEN SHAFT	-	PUR	0.00	0.113	0.11	1	2.00	2.00
042	OIL SEAL	FRONT GAS GEN SHAFT	-	PUR	0.00	0.106	0.11	1	0.10	0.10
044	SCREW	-	-	PUR	0.00	0.006	0.05	8	0.01	0.08
045	HOUSING	BEARING	NOD. IRON	PUR	0.00	0.745	0.74	1	0.50	0.50
046	INSULATION	-	-	PUR	0.20	1.045	1.05	1	0.21	0.21
047	O RING	COMP DIFFUSER	-	PUR	0.00	0.068	0.14	2	0.03	0.06
049	INSULATION	-	-	PUR	0.20	1.576	1.58	1	0.32	0.32
051	SCREW	-	-	PUR	0.00	0.006	0.07	12	0.01	0.12
052	GASKET	-	PAPER	PUR	0.00	0.081	0.08	1	0.10	0.10
053	NUT	-	-	PUR	0.00	0.006	0.07	12	0.01	0.12
055	INSULATION	-	-	PUR	0.20	2.090	2.09	1	0.42	0.42
059C	RECUPERATOR	(CORE)	CERVIT	PUR	0.00	21.728	43.46	2	30.00	60.00
060	DOWEL PIN	-	-	PUR	0.00	0.002	0.00	1	0.01	0.01
061	O RING	-	-	PUR	0.00	0.104	0.10	1	0.03	0.03
062	INSULATION	-	-	PUR	0.20	4.440	4.44	1	0.89	0.89

Table A-26 (Cont'd.)

AUTOMOTIVE GAS TURBINE SELECTION STUDY
PARTS LIST
PD-2 ENGINE (FREE TURBINE)

PART	NAME	DESCRIPTION	MATERIAL	TYPE	COST/LB	WEIGHT	TOTAL WEIGHT	NUMBER OF PARTS	COST PER PART	TOTAL COST
064	HOUSING	BEARING	NOD. IRON	PUR	0.00	41.549	1.55	1	0.50	0.50
065	SEAL	METALLIC	HS-25	PUR	0.00	0.053	0.05	1	0.25	0.25
066	RING SEAL	GAS GEN TURBINE	HS-25	PUR	0.00	0.028	0.08	3	0.15	0.45
069	BEARING	REAR GAS GEN	-	PUR	0.00	0.196	0.20	1	2.00	2.00
070	OIL SEAL	GAS GEN TURBINE	-	PUR	0.00	0.204	0.20	1	0.10	0.10
071	SNAP RING	GAS GEN TURB OIL SEAL	-	PUR	0.00	0.017	0.02	1	0.04	0.04
072	SCREW	-	-	PUR	0.00	0.003	0.02	8	0.01	0.08
073	RING SEAL	POWER TURBINE	302 SST	PUR	0.00	0.023	0.07	3	0.10	0.30
074	OIL SEAL	POWER TURBINE	-	PUR	0.00	0.076	0.08	1	0.10	0.10
076	GASKET	SEAL	PAPER	PUR	0.00	0.039	0.04	1	0.10	0.10
077	BOLT	-	304 SST	PUR	0.00	0.034	0.27	8	0.01	0.08
078	NUT	-	304 SST	PUR	0.00	0.004	0.05	12	0.01	0.12
079	SCREW	-	304 SST	PUR	0.00	0.004	0.05	12	0.01	0.12
081	INSULATION	-	-	PUR	0.20	4.739	4.74	1	0.95	0.95
085	BOLTS	-	4140 STEEL	PUR	0.00	0.007	0.21	30	0.01	0.30
086	INSULATION	POWER TURB BEAR HOUSING	-	PUR	0.20	3.890	3.89	1	0.78	0.78
088	OIL PUMP	-	-	PUR	0.00	1.269	1.27	1	2.15	2.15
090	BOLTS	-	-	PUR	0.00	0.002	0.02	8	0.01	0.08
091	WOODRUFF KEY	OIL PUMP DRIVE SHAFT	-	PUR	0.00	0.001	0.00	1	0.03	0.03
092	SNAP RING	OIL PUMP BEAR OUTER	-	PUR	0.00	0.018	0.02	1	0.04	0.04
093	SNAP RING	OIL PUMP BEAR INNER	-	PUR	0.00	0.009	0.02	2	0.04	0.08
094	BEARING	OIL PUMP	-	PUR	0.00	0.339	0.68	2	2.80	5.60
096	BOLT	-	-	PUR	0.00	0.002	0.05	24	0.01	0.24
099	BOLT	-	-	PUR	0.00	0.017	0.41	24	0.01	0.24
104	INSULATION	-	-	PUR	0.20	1.125	1.13	1	0.22	0.22
105	NUT	-	-	PUR	0.00	0.012	0.10	8	0.01	0.08
107	BOLT	-	304 SST	PUR	0.00	0.012	0.10	8	0.01	0.08
109	NUT	-	304 SST	PUR	0.00	0.003	0.04	12	0.01	0.12
110	SCREW	-	304 SST	PUR	0.00	0.003	0.04	12	0.01	0.12
111	RING SEAL	EXHAUST SCROLL	INC. X	PUR	0.00	0.059	0.18	3	0.20	0.60
113	SNAP RING	POWER TURB REAR BRG	-	PUR	0.00	0.022	0.02	1	0.04	0.04
114	O RING	OIL PUMP	NEOPRENE	PUR	0.00	0.016	0.03	2	0.03	0.06
116	OIL SEAL	OIL PUMP SHAFT REAR BRG	-	PUR	0.00	0.018	0.02	1	0.10	0.10

Table A-26 (Cont'd.)

AUTOMOTIVE GAS TURBINE SELECTION STUDY
PARTS LIST
PD-2 ENGINE (FREE TURBINE)

PART	NAME	DESCRIPTION	MATERIAL	TYPE	COST/LB	WEIGHT	TOTAL WEIGHT	NUMBER OF PARTS	COST PER PART	TOTAL COST
117	LOCK WASHER	OIL PUMP SHAFT	-	PUR	0.00	0.007	0.01	1	0.04	0.04
118	NUT	OIL PUMP SHAFT	-	PUR	0.00	0.022	0.02	1	0.01	0.01
120	MARMAN CLAMP	-	-	PUR	0.00	0.457	0.46	1	1.00	1.00
121	SEAL RING	MARMAN FLANGE	-	PUR	0.00	0.023	0.02	1	0.10	0.10
123	O RING	-	NEOPRENE	PUR	0.00	0.212	0.21	1	0.03	0.03
124	BOLT	-	-	PUR	0.00	0.049	1.18	24	0.01	0.24
125	O RING	-	NEOPRENE	PUR	0.00	0.003	0.00	1	0.03	0.03
127	OIL PUMP	SCAVENGE	-	PUR	0.00	0.255	0.25	1	0.55	0.55
129	SCREWS	-	-	PUR	0.00	0.001	0.01	8	0.01	0.08
130	WOODRUFF KEY	SCAVENGE OIL PUMP SHAFT	-	PUR	0.00	0.000	0.00	1	0.03	0.03
131	GASKET	-	-	PUR	0.00	0.023	0.05	2	0.10	0.20
132	MARMAN CLAMP	-	-	PUR	0.00	0.274	0.55	2	1.00	2.00
134	LOCK WASHER	-	-	PUR	0.00	0.001	0.05	60	0.01	0.30
135	SCREW	-	-	PUR	0.00	0.010	0.60	60	0.01	0.30
137	ACTUATOR	-	-	PUR	0.00	1.524	3.05	2	2.00	4.00
138	SCREW	-	-	PUR	0.00	0.008	0.06	8	0.01	0.08
139	O RING	-	NEOPRENE	PUR	0.00	0.005	0.01	2	0.03	0.06
140	O RING	-	NEOPRENE	PUR	0.00	0.002	0.00	2	0.03	0.06
141	BRACKET	-	1010 STEEL	PUR	0.00	0.509	1.02	2	0.10	0.20
142	BOLT	-	-	PUR	0.00	0.006	0.05	8	0.01	0.08
143	BRACKET	-	1010 STEEL	PUR	0.00	0.569	1.14	2	0.10	0.20
144	NUT	-	-	PUR	0.00	0.008	0.06	8	0.01	0.08
145	SCREW	SOC HEAD	-	PUR	0.00	0.106	0.32	3	0.01	0.03
148	MARMAN CLAMP	-	-	PUR	0.00	0.231	0.46	2	1.00	2.00
150	FITTING	TUBE	AL	PUR	0.00	0.009	0.03	3	0.05	0.15
152	BUSHING	ADAPTER	AL	PUR	0.00	0.035	0.07	2	0.01	0.02
153	ELBOW FITTING	-	AL	PUR	0.00	0.019	0.02	1	0.05	0.05
154	LOCK WASHER	-	-	PUR	0.00	0.008	0.01	1	0.01	0.01

TOTAL WEIGHT = 83.33 LBS. TOTAL COST = $122.64 NUMBER OF PARTS = 484

Table A-27

Miscellaneous

AUTOMOTIVE GAS TURBINE SELECTION STUDY
PARTS LIST
PD-2 ENGINE (FREE TURBINE)

PART	NAME	DESCRIPTION	MATERIAL	TYPE	COST/LB	WEIGHT	TOTAL WEIGHT	NUMBER OF PARTS	COST PER PART	TOTAL COST
067	TIP SEAL	GAS GEN TURBINE	HAST. X	HNC	70.80	0.122	0.12	1	8.34	8.64
075	TIP SEAL	POWER TURBINE	304 SST	HNC	10.00	0.130	0.13	1	1.30	1.30
083	RING SEAL	GAS GEN NOZZLE HOUSING	HS-25	-	0.00	0.048	0.14	3	0.15	0.45
119	FLANGE	MARMAN	-	-	0.00	0.000	0.00	0	0.00	0.00
122	MARMAN FLANGE	-	-	-	0.00	0.000	0.00	0	0.00	0.00
149	GASKET	-	-	-	0.00	0.000	0.00	2	0.00	0.00

TOTAL WEIGHT = 0.40 LBS. TOTAL COST = $ 10.39 NUMBER OF PARTS = 7

Table A-28

Hastelloy X and Inconel 713LC

AUTOMOTIVE GAS TURBINE SELECTION STUDY
PARTS LIST
PD-2 ENGINE (FREE TURBINE)

PART	NAME	DESCRIPTION	MATERIAL	TYPE	COST/LB	WEIGHT	TOTAL WEIGHT	NUMBER OF PARTS	COST PER PART	TOTAL COST
009	TURBINE	GAS GEN	INC 713LC	PIC	6.50	2.230	2.23	1	14.50	14.50
057	CAP	FUEL INJECTOR	HAST. X	CST	10.00	0.195	0.20	1	1.95	1.95
063	LINER	COMBUSTOR	HAST. X	EXT	3.50	0.751	0.75	1	2.63	2.63
067	TIP SEAL	GAS GEN TURBINE	HAST. X	HNC	70.50	0.122	0.12	1	8.64	8.64
080A	NOZZLE HOUSING	GAS GEN TURBINE	HAST. X	CST	10.00	6.832	6.83	1	68.32	68.32
080B	NOZZLE HOUSING	GAS GEN TURBINE	HAST. X	EXT	3.50	0.580	0.58	1	2.03	2.03

TOTAL WEIGHT = 10.71 LBS. TOTAL COST = $ 98.06 NUMBER OF PARTS = 6

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Table A-29

Steel

AUTOMOTIVE GAS TURBINE SELECTION STUDY
PARTS LIST
PD-2 ENGINE (FREE TURBINE)

PART	NAME	DESCRIPTION	MATERIAL	TYPE	COST/LB	WEIGHT	TOTAL WEIGHT	NUMBER OF PARTS	COST PER PART	TOTAL COST
002	PIN	COMP BULLETNOSE	4140 STEEL	PUR	0.00	0.004	0.00	1	0.01	0.01
003	NUT	COMPRESSOR	4140 STEEL	BAR	0.20	0.089	0.09	1	0.02	0.02
004	SHAFT	GAS GENERATOR	4340 STEEL	BAR	0.20	1.372	1.37	1	0.27	0.27
006	WOODRUFF KEY	COMPRESSOR	4140 STEEL	PUR	0.00	0.011	0.01	1	0.03	0.03
007	SEAL RING	COMPRESSOR	4140 STEEL	BAR	0.20	0.050	0.05	1	0.01	0.01
008	GEAR	GAS GEN SHAFT	8620 STEEL	FOR	0.30	0.153	0.15	1	0.05	0.05
010	GEAR	STARTER GAS GEN	8620 STEEL	FOR	0.30	0.300	0.30	1	0.09	0.09
013	GEAR SHAFT	STARTER GAS GEN	4340 STEEL	BAR	0.20	0.340	0.34	1	0.07	0.07
014	GEAR	STARTER GAS GEN	8620 STEEL	FOR	0.30	0.052	0.05	1	0.02	0.02
016	GEAR	STARTER	8620 STEEL	FOR	0.30	0.205	0.21	1	0.06	0.06
023	SHAFT	POWER TURBINE	4340 STEEL	BAR	0.20	1.257	1.26	1	0.25	0.25
024	SEAL RING	POWER TURBINE SHAFT	4140 STEEL	BAR	0.20	0.070	0.07	1	0.01	0.01
026	SPACER	POWER TURB SHAFT BEARING	4140 STEEL	TUB	0.20	0.893	0.89	1	0.18	0.18
027	RETAINER	AFT POWER TURB SHAFT BEAR	4140 STEEL	BAR	0.20	0.052	0.05	1	0.01	0.01
028	GEAR	POWER TURB DRIVE	8620 STEEL	FOR	0.30	0.122	0.12	1	0.04	0.04
030	NUT	POWER TURBINE SHAFT	4140 STEEL	BAR	0.20	0.027	0.03	1	0.01	0.01
031	GEAR ASSY	REDUCTION	8620 STEEL	FOR	0.30	2.680	2.68	1	0.80	0.80
034	GEAR ASSY	ACCESSORY DRIVE	8620 STEEL	FOR	0.30	4.230	4.23	1	1.27	1.27
041	SPACER	GAS GEN GEAR SHAFT BRGS	4140 STEEL	TUB	0.20	0.170	0.17	1	0.03	0.03
043	RETAINER	GAS GEN SHAFT FNT OIL SEAL	4140 STEEL	BAR	0.20	0.141	0.14	1	0.03	0.03
059B	RECUPERATOR	(EXHAUST PIPE)	AL. STEEL	SHT	0.20	7.080	14.16	2	1.42	2.83
085	BOLTS	-	4140 STEEL	PUR	0.00	0.007	0.21	30	0.01	0.30
095	DRIVE SHAFT	OIL PUMP	4140 STEEL	FOR	0.30	1.596	1.60	1	0.48	0.48
141	BRACKET	-	1010 STEEL	PUR	0.00	0.509	1.02	2	0.10	0.20
143	BRACKET	-	1010 STEEL	PUR	0.00	0.569	1.14	2	0.10	0.20

TOTAL WEIGHT = 30.34 LBS. TOTAL COST = $ 7.27 NUMBER OF PARTS = 57

Table A-30

Nodular Iron

AUTOMOTIVE GAS TURBINE SELECTION STUDY
PARTS LIST
PD-2 ENGINE (FREE TURBINE)

PART	NAME	DESCRIPTION	MATERIAL	TYPE	COST/LB	WEIGHT	TOTAL WEIGHT	NUMBER OF PARTS	COST PER PART	TOTAL COST
039	SHEAVE	V BELT	NOD. IRON	PUR	0.00	0.477	1.43	3	0.26	0.78
045	HOUSING	BEARING	NOD. IRON	PUR	0.00	0.745	0.74	1	0.50	0.50
048	DIFFUSER	COMPRESSOR	NOD. IRON	CST	0.30	8.726	8.73	1	2.62	2.62
050	COVER PLATE	FRONT	NOD. IRON	CST	0.25	13.985	18.98	1	4.75	4.75
054	HOUSING	BEARING GAS GEN	NOD. IRON	CST	0.25	41.384	41.38	1	10.35	10.35
058	COVER	RECUPERATOR TOP	NOD. IRON	CST	0.30	7.533	15.07	2	2.26	4.52
064	HOUSING	BEARING	NOD. IRON	PUR	0.00	1.549	1.55	1	0.50	0.50
087	HOUSING	BEARING POWER TURBINE	NOD. IRON	CST	0.25	28.776	28.78	1	7.19	7.19
089	COVER	OIL PUMP	NOD. IRON	CST	0.30	0.517	0.52	1	0.16	0.16
097	COVER	REDUCTION GEAR BEARING	NOD. IRON	CST	0.30	0.430	0.43	1	0.13	0.13
098	COVER	GEAR BOX	NOD. IRON	CST	0.25	36.632	36.63	1	9.16	9.16
100	COVER	ACCESSORY DRIVE SHAFT REAR BEAR	NOD. IRON	CST	0.30	1.217	1.22	1	0.37	0.37
106	HOUSING	EXHAUST SCROLL RING SEAL	NOD. IRON	CST	0.30	3.833	3.83	1	1.15	1.15
115	COVER	OIL PUMP SHAFT REAR BRG	NOD. IRON	CST	0.30	1.217	1.22	1	0.37	0.37
126	COVER	BOTTOM RECUP	NOD. IRON	CST	0.30	7.533	15.07	2	2.26	4.52
128	COVER	SCAVENGE OIL PUMP	NOD. IRON	CST	0.30	0.051	0.05	1	0.02	0.02

TOTAL WEIGHT = 175.63 LBS. TOTAL COST = $ 47.06 NUMBER OF PARTS = 20

Table A-31

304 Stainless Steel

AUTOMOTIVE GAS TURBINE SELECTION STUDY
PARTS LIST
PD-2 ENGINE (FREE TURBINE)

PART	NAME	DESCRIPTION	MATERIAL	TYPE	COST/LB	WEIGHT	TOTAL WEIGHT	NUMBER OF PARTS	COST PER PART	TOTAL COST
056	OUTER SHELL	COMBUSTOR	304 SST	CST	2.00	7.213	7.21	1	14.43	14.43
059A	RECUPERATOR	(SHELL)	304 SST	SHT	0.70	26.172	52.34	2	18.32	36.64
068	CASING	TURBINE INNER	304 SST	CST	2.00	2.891	2.89	1	5.78	5.78
075	TIP SEAL	POWER TURBINE	304 SST	HNC	10.00	0.130	0.13	1	1.30	1.30
077	BOLT	-	304 SST	PUR	0.00	0.034	0.27	8	0.01	0.08
078	NUT	-	304 SST	PUR	0.00	0.004	0.05	12	0.01	0.12
079	SCREW	-	304 SST	PUR	0.00	0.004	0.05	12	0.01	0.12
101	ACTUATOR LINK	POWER TURB NOZZLE	304 SST	CST	2.00	0.049	1.76	36	0.10	3.53
102	ACTUATOR RING	POWER TURB NOZZLE	304 SST	CST	2.00	2.240	2.24	1	4.48	4.48
103	CASING	POWER TURB OUTER	304 SST	CST	2.00	11.481	11.48	1	22.96	22.96
107	BOLT	-	304 SST	PUR	0.00	0.012	0.10	8	0.01	0.08
108	SCROLL	EXHAUST	304 SST	CST	2.00	21.702	21.70	1	43.40	43.40
109	NUT	-	304 SST	PUR	0.00	0.003	0.04	12	0.01	0.12
110	SCREW	-	304 SST	PUR	0.00	0.003	0.04	12	0.01	0.12
112	STRUT	TURBINE EXHAUST	304 SST	CST	2.00	0.191	0.76	4	0.38	1.53
147	TUBE	STRUT	304 SST	CST	2.00	0.375	1.50	4	0.75	3.00

TOTAL WEIGHT = 102.56 LBS.   TOTAL COST = $137.69   NUMBER OF PARTS = 116

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Table A-32

Aluminum

AUTOMOTIVE GAS TURBINE SELECTION STUDY
PARTS LIST
PD-2 ENGINE (FREE TURBINE)

PART	NAME	DESCRIPTION	MATERIAL	TYPE	COST/LB	WEIGHT	TOTAL WEIGHT	NUMBER OF PARTS	COST PER PART	TOTAL COST
001	BULLETNOSE	COMPRESSOR	355 AL	CST	0.45	0.149	0.15	1	0.07	0.07
150	FITTING	TUBE	AL	PUR	0.00	0.009	0.03	3	0.05	0.15
151	TUBE	1/4 IN OD	AL	EXT	0.40	0.028	0.03	1	0.01	0.01
152	BUSHING -	ADAPTER	AL	PUR	0.00	0.035	0.07	2	0.01	0.02
153	ELBOW FITTING-		AL	PUR	0.00	0.019	0.02	1	0.05	0.05

TOTAL WEIGHT = 0.29 LBS. TOTAL COST = $ 0.30 NUMBER OF PARTS = 8

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Table A-33

Precision Investment Castings

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                                AUTOMOTIVE GAS TURBINE SELECTION STUDY
                                        PARTS LIST
                                       PD-2 ENGINE (FREE TURBINE)

PART  NAME         DESCRIPTION               MATERIAL    TYPE COST/LB WEIGHT  TOTAL   NUMBER     COST     TOTAL
                                                                              WEIGHT  OF PARTS  PER PART   COST

009  TURBINE       GAS GEN                   INC 713LC    PIC   6.50   2.230   2.23       1      14.50     14.50
022  TURBINE       POWER                     CMR-60       PIC   2.77   2.061   2.06       1       5.71      5.71

TOTAL WEIGHT =    4.29 LBS.   TOTAL COST = $ 20.20     NUMBER OF PARTS =   2
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